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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOL. 34 MAY 1912 NUMBER 5

THE SPRING MEETING

The program for the Spring Meeting at Cleveland is published herewith, and although subject to change is substantially in the form in which the professional sessions and the entertainment of the members and their friends will be carried out. It should be noted especially that the date of the meeting is May 28–31 instead of during the first week in June as first announced. The attractiveness of Cleveland as a city and its location on the lake, together with its remarkable industrial development, all contribute to make it one of the most interesting places for a meeting of engineers. The generous hospitality which its residents have shown on many occasions is known to all and the efforts which the local membership are putting forth to insure that the time of every visiting member shall be fully and pleasantly occupied may well lead to great anticipations on the part of those expecting to attend. The arrangements for the reception of the visitors are in the hands of an Executive Committee of which Past-President Ambrose Swasey is Chairman; Professor Robert H. Fernald, Vice-Chairman; F. W. Ballard, Secretary and R. B. Sheridan, Treasurer. It requires but a glance at the program to show how profitable and enjoyable an occasion this Committee and the Committee on Meetings of the Society have arranged.

TENTATIVE PROGRAM

TUESDAY, MAY 28

Registration at headquarters, Chamber of Commerce Hall, after 10 a.m.

Informal reception at the home of Mr. and Mrs. Ambrose Swasey, 7808 Euclid Ave., from 4 to 6 p.m.

Membership reunion and informal evening, Chamber of Commerce Hall, 8.30 p.m.

WEDNESDAY, MAY 29

Business Meeting, 10 a.m., Chamber of Commerce Hall
Professional Session, 10.30 a.m., Chamber of Commerce Hall

A New Analysis of the Cylinder Performance of Reciprocating Engines, J. Paul Clayton

Equipment of a Modern Flour Mill on a Gradual Reduction System, John F. Harrison and W. W. Nichols

Design and Mechanical Features of the California Gold Dredge, Robert E. Cranston (To be read by title, discussion invited)

Simultaneous Session, Gas Power Section, 10.30 a.m., Chamber of Commerce Library

Problems in Natural Gas Engineering, Thomas R. Weymouth

Other papers to be announced.

Entertainment

Special Inspection Trip for ladies to plant of H. Black Company, cloak manufacturers, 10 a.m.

Lunch at Chamber of Commerce Hall, 12.30 p.m.

Inspection of local manufacturing and power plants, 2 p.m.

Automobile trip for ladies through the parks, 2 p.m.

Tea at Country Club, for members and guests, 4 to 6 p.m.

Lecture, Chamber of Commerce Hall, 8.30 p.m.

Sound Waves, How to Photograph Them and What They Mean, Dr. Dayton C. Miller, Case School of Applied Science

The lecture will be illustrated with apparatus and experiments and especially with the new instrument, the "Phonodeik," for projecting sound waves directly upon the screen.

THURSDAY, MAY 30

Professional Session, Chamber of Commerce Hall, 10 a.m.

New Processes for Chilling Cast Iron, Thos. D. West

STRENGTH OF STEEL TUBES, PIPES AND CYLINDERS UNDER INTERNAL FLUID PRESSURE, Reid T. Stewart

ON THE CONTROL OF SURGES IN WATER CONDUITS, W. F. Durand

SPEED REGULATION IN HYDRO-ELECTRIC PLANTS, Wm. F. Uhl

(To be read by title, discussion invited)

Entertainment

Excursion on Lake Erie, 1 to 5 p.m.

Reception and dance at Colonial Club, 8.30 p.m.

FRIDAY, MAY 31

Professional Session, 10 a.m., Chamber of Commerce Hall

THE PRESENT STATE OF DEVELOPMENT OF LARGE STEAM TURBINES, A. G. Christie

A DISCUSSION OF CERTAIN THERMAL PROPERTIES OF STEAM, G. A. Goodenough

THE REDUCTION IN TEMPERATURE OF CONDENSING WATER RESERVOIRS DUE TO COOLING EFFECT OF AIR AND EVAPORATION, W. B. Ruggles

RESULTS OF TESTS ON THE DISCHARGE CAPACITY OF SAFETY VALVES, E. F. Miller and A. B. Carhart (To be read by title, discussion invited)

Entertainment

Lunch at Chamber of Commerce Hall, 12 o'clock sharp.

Excursion to Akron, leaving Chamber of Commerce, 12.30 p.m.

Inspection of plant of The Goodrich Rubber Company.

Inspection of plant of Wellman-Seaver-Morgan Company.

Inspection of plant of Diamond Match Company at Barberton.

Ladies are invited to make the trip to Akron.

RAILROAD TRANSPORTATION NOTICE

Every effort has been made to secure reduced rates on the certificate plan for members and guests in attendance at the Spring Meeting, but the widespread adoption of the two-cent rate and similar restrictions have operated to make this impossible.

All the railroads will provide special cars or special trains if the advance reservations warrant such service. Members are therefore urged to secure tickets and Pullman reservations a day or two in advance, in all cases stating that accommodations are desired on The American Society of Mechanical Engineers' special.

TRAIN SCHEDULES[1]

New York Central Lines

Lv. New York	5.30 p.m.	6.30 p.m.	8.02 p.m.
Lv. Boston	2.00	2.00	4.50 p.m.
Lv. Worcester	3.12	3.12	6.00 p.m.
Lv. Providence			4.10 p.m.
Lv. Albany	8.15	9.55	11.45 p.m.
Arr. Cleveland	7.15 a.m.	7.30 a.m.	10.47 a.m.
Fare New York to Cleveland	$14	$15	$13

Pennsylvania

Lv. New York	5.04 p.m.	6.32 p.m.		8.30 p.m.
Lv. North Philadelphia	7.02 p.m.	8.17 p.m		
Lv. Philadelphia				10.48 p.m.
Lv. Washington			6.45 p.m.	
Lv. Baltimore			7.52 p.m.	
Lv. Harrisburg	9.39 p.m.	10.34 p.m.	10.17 p.m.	1.20 a.m.
Arr. Cleveland	7.30 a.m.	7.30 a.m.	7.30 a.m.	11.30 a.m.

Lackawanna-Nickel Plate

Lv. New York	6.30 p.m.
Lv. Scranton	11.28 p.m.
Arr. Cleveland	11.19 a.m.
Fare New York to Cleveland	$12

PULLMAN RESERVATIONS

Pullman rates on all roads are: Lower berth, $3; upper berth, $2.40; drawing room, $11. The Society cannot undertake to arrange transportation matters for members. Tickets and Pullman reservations may be obtained at any ticket office or by mail from the following: NEW YORK CENTRAL LINES: New York, Mr. W. V. Lifsey, G. E. P. A., Broadway and 30th St.; Boston, Mr. C. E. Colony, C. P. A., Boston & Albany R.R.; PENNSYLVANIA: New York, Mr.C. Studds, D. P. A., 263 Fifth Ave.; Philadelphia, Mr. R. L. Stall, D. P. A., 1433 Chestnut St.; Baltimore, Mr. Hugh Hasson, D. P. A., Baltimore and Calvert Sts.; Washington, Mr. B. M. Newbold, D. P. A., 15th and G Sts.; LACKAWANNA-NICKEL PLATE: NewYork, Mr. A. W. Ecclestone, D. P. A., Nickel Plate Road, 385 Broadway.

[1] Subject to change.

HOTEL RATES FOR SPRING MEETING AT CLEVELAND

	Without Bath			With Bath		
	Single Room	Single Room, Two Persons	Double Room, Two Persons	Single Room	Single Room, Two Persons	Double Room Two Persons
Hollenden Hotel	$2.00	$3.00	$3.50	$2.50 to 4.00	$4.00 to 5.00	$5.00 to 7.00
The Colonial	2.00	.....	.....	2.50 to 5.00	.....	.....
Hotel Euclid	1.50	.....	.....	2.00 to 5.00	.....	.....

GERMAN MUSEUM COMMISSION

MEETING IN NEW YORK, APRIL 9.

At a meeting in New York on April 9, the greetings of the Society were extended to the members of the German Museum Commission, now on a visit to America to study important engineering achievements. The Commission, which is headed by Dr. Oscar von Miller, member of the House of Lords of Bavaria and President of the Verein deutscher Ingenieure, includes a number of prominent scientific men, His Excellency Dr. Count von Podewils-Durniz, former Secretary of State of Bavaria; Dr. W. von Borscht, Lord Mayor of Munich and Privy Councillor; Dr. W. von Dyck, Privy Councillor, Rector Emeritus of the Technical University of Munich; Herr Ph. Gelius, Architect of the Museum; Herr Alex. Schirmann, Director of the Library of the Museum; Dr. Fuchs, Professor of Physics and Mathematics; Engineer Fr. Orth, Mining Engineer, Director of Sections of the Museum; Engineer Kurt Trautwein, Civil Engineer, Director of Divisions of the Museum; and Dr. Colin Ross, Secretary of the Commission.

Dr. Alex. C. Humphreys, President of the Society, presided and spoke of the splendid work of the Museum for science. The speaker of the evening, Dr. Von Miller, was then introduced, and his address read by Charles Whiting Baker.

The purpose for which the German Museum was built, he said, was to show by actual samples and models the development of the natural and technical sciences, in an instructive and attractive form. In this way the layman receives concise and impressive instruction in all important departments of natural science and technology, while the numerous historical originals give the expert an opportunity for research in the domain of his art.

The great inventions are presented in the Museum by a series of constructions, from the first crude designs, through the successive stages of development to the most recently perfected devices, thus showing how each new construction depends upon former achievements, and how these again become the point of departure for

further advancement and improvement. Whenever possible, this is done by the original apparatus used or constructed by the inventor himself. When the original apparatus could not be obtained, reproductions were made for the Museum; thus in the case of the "Old Puffing Billy," of which the original is in the South Kensington Museum, a reproduction was made, so exact in the smallest details even to rust spots, that the photographs of the original and of the reproduction cannot be distinguished from each other.

When an object cannot be exhibited in the original on account of its dimensions, plans and models are included in the Museum collections. Thus the development of the human dwelling, from the prehistoric cave to the modern skyscraper, is shown by a long series of models, arranged so that relative importance is prominently brought out by contrasting different building systems.

The most valuable and beautiful originals would, however, often remain unintelligible to laymen if they were not explained more fully. The sectional models are the first step in this system of technical education. Some of the models are only sectionally cut and uncovered, in order to explain to the observer the interior arrangement, while in others ways are devised to show also the particular action on which the machine is based. For example, in Alban's tubular boiler the way in which water and steam move and the gases of combustion return are illustrated by arrow marks, while in a Krupp converter the rising air bubbles are painted on glass. Some of the originals are supplemented by working models showing the mode of operation of the machines. Such working models are used not only in the machine section, but also in the natural science departments, where every visitor can in this way perform some of the experiments for himself. There is, for example, an apparatus for demonstrating the properties of liquid air. Five vessels filled with it can be moved up and down in a cabinet to dip substances in it, experimentally, and the visitor can see for himself how mercury freezes in liquid air, rubber loses its elasticity, carbon dioxide gas becomes solid, and the electric resistance of a supply wire of an electric lamp falls off. In an X-ray cabinet a switch is moved by opening the door; the cabinet is darkened, and the switching in of the X-ray tubes follows automatically, and a screen of chloride of barium is illuminated so that an object selected by the observer, himself, such as his hand or purse, can be seen illuminated by the X-rays. After two minutes the X-ray tubes are switched off and the cabinet automatically lighted again.

Whenever necessary, descriptions, drawings and concise explanation of processes are placed at the beginning of many groups or above particular exhibits. Statistical tables and tabulated statements are also included to show the influence of science and technic on human culture and economic life, the relations of production and consumption, market conditions, etc.

The main objects of the Museum, illustration and instruction, are accomplished in the best and most attractive manner possible, considerable stress being laid on the artistic point of view. Each object is, as far as possible, presented in a proper setting, in a room adapted to it. Thus various old blacksmith's tools have found a place in an old smith's shop; the chemical apparatus and vessels used at different epochs are exhibited in a series of laboratories, from the quaint laboratory of the alchemist and phlogistian to the familiar types of today.

The needs of the expert as well as the layman are fully cared for also. A library connected with the Museum embraces the entire literature of natural science and technology of the present day, as well as numerous historical and rare works, and letters of celebrated men of science. An extensive collection of plans affords to the engineer and architect an opportunity to acquaint himself with the way in which others solved the questions which interest him.

A Hall of Fame serves to commemorate great investigators and technicians, whose names should be as familiar to the public as those of great statesmen and generals.

A new home is now being built for the Museum which will provide about 350,000 sq. ft. of space for exhibition purposes, not including the large library, reading rooms, lecture rooms and meeting halls. There will also be special facilities for experimental work, such as a tower 230 feet high for geodetical and meteorological observations, and a station for wireless telegraphy and telephony.

Following the reading of this paper, Dr. von Miller thanked the Society in his native tongue for the assistance rendered the Commission and expressed the hope that should the American people build a similar museum they would come over to Germany and learn how not to repeat some of their mistakes.

Dr. W. von Dyck then spoke of the publication of biographies and documents of science and technic which the Museum is planning, and presented to the President of the Society the first volume of the series, written by himself, containing the biography of Georg von

Reichenbach, an engineer and astronomer, who lived at the time when Watt was engaged in laying the foundations of modern engineering. This was accepted by Dr. Humphreys and has been placed in the Library of the Society.

A reception in the rooms of the Society followed the meeting, in which all were given an opportunity to greet the Commissioners personally.

VISIT TO PHILADELPHIA

A committee of the Society greeted the Commission upon their arrival in Philadelphia on April 11 and assisted them in their inspection of various interesting points during their two days' visit. Representatives of the Committee, which consisted of A. C. Jackson, Chairman, D. R. Yarnall, Charles Day, Morris L. Cooke, T. C. McBride, S. L. Kneass, James M. Dodge, W. C. Kerr, S. S. Webster, Coleman Sellers, Jr., and J. E. Gibson, met the Commissioners at the train and escorted them to the office of the Mayor, who received them most cordially. During the morning the City planning exhibit in the City Hall, Wanamaker's store, and the Girard Trust Company were inspected and luncheon was served at the Engineers Club. The Baldwin Locomotive Works, the new William Penn High School, and the Boys' Central High School occupied the remainder of the day. On Friday, April 12, trips were made to the University of Pennsylvania, where the new engineering building and the university museum proved especially interesting; the School of Industrial Art, Franklin Institute, Memorial Hall in Fairmount Park, and the J. G. Johnson Art Gallery. The Mayor entertained the Commission at luncheon on Friday in the Union League Club.

ITINERARY

The itinerary of the Commission as planned includes Washington, Pittsburgh, Chicago, Niagara Falls and Buffalo, Schenectady, Boston, and a return visit to New York, where they will sail for Europe on April 30. Committees of the Society have been formed at each of these points to greet the Commission and to assist them in making the tours of inspection which they desire. Gen. William H. Bixby acted as chairman of the reception committee in Washington, George M. Brill, Vice-President, in Chicago, C. H. Bierbaum in Buffalo, H. G. Reist in Schenectady, and Prof. I. N. Hollis in Boston.

GENERAL NOTES

MEETINGS IN HONOR OF DR. DIESEL

Dr. Rudolph Diesel, inventor of the Diesel engine and director of the Verein deutscher Ingenieure, is expected to be in New York on April 30, at which time Honorary Membership will be conferred upon him by the Society at a meeting in the auditorium of the Engineering Societies Building. Dr. Diesel is widely known to engineers and the public at large through the fame of his invention, and members of sister societies have been invited to participate in this meeting in his honor. Following the conferring of Honorary Membership, Dr. Diesel will give an illustrated address, under the auspices of the Gas Power Section of the Society, on the Development of the Diesel Engine.

On April 13 Dr. Diesel made a brief address upon the same subject before a large audience in St. Louis, and during his stay in that city was tendered a dinner at the Mercantile Club by the Associated Engineering Societies. On April 17 he visited Cornell University, making a similar address, and on April 26 spoke at the United States Naval Academy at Annapolis upon the Relation of the Gas Engine to the Propulsion of Vessels. His complete and more carefully prepared address upon the Diesel Engine will be made at the larger gathering in New York.

MEASUREMENT OF TAPS, BOLTS AND SCREWS

As a result of interest awakened in the subject of the measurement of taps, bolts and screws at the March meeting in New York, it has been suggested by the chairman of the Committee on Meetings in New York that the Society should appoint a committee to make recommendations for standard methods of measurement. It is probable that comparatively few users of these articles have definite knowledge of the degree of accuracy which they are securing when purchasing, and few even have suitable measuring instruments for screw threads. Much delay is occasioned in assembling work through the lack of proper methods and instruments for measuring the screws

and taps that are to be used. It has further been suggested that the scope of the committee might be extended to include the standardization of limits for other machine parts as well; for it is only by such standardization in any manufacturing plant that interchangeability can be attained and the cost of assembling brought within a reasonable figure, not to mention the possibility of duplicating work on subsequent orders.

PATENT CONFERENCE AT WASHINGTON

Upon invitation of the Patent Law Association, an informal conference has been held in Washington, April 15 and 16, to discuss the situation resulting from several bills now pending in Congress relating to proposed changes in the patent laws. Several of the national societies participated and to represent The American Society of Mechanical Engineers the Council appointed as Honorary Vice-Presidents, W. H. Blauvelt, consulting engineer of the Semet-Solvay Company, Syracuse, N. Y.; B. F. Wood, assistant engineer of the motive power department, Pennsylvania Railroad Company, Altoona, Pa.; and J. B. F. Herreshoff, president and treasurer of the Herreshoff Manufacturing Company, Bristol, R. I.

In appointing these representatives, the Society believed that so many of the membership had large interests at stake in the patent system that it should be ably represented in any conference relating to the matter of patent law, and that the Society at large should take some part in an affair affecting so important a feature of national development.

TESTIMONIAL FROM STEVENS ALUMNI TO DR. DENTON

Dr. James E. Denton, a charter member of the Society and one of its managers from 1889–1892, now emeritus professor of engineering practice of Stevens Institute of Technology, was recently honored by the alumni of that college. Dr. Denton, who is himself a graduate of Stevens, had long been associated with its faculty, but was recently obliged by impaired health to retire from active work. As a tribute to the mechanical genius and strong personality of Dr. Denton, a handsome watch and fob, accompanied by a beautifully engraved testimonial, bearing the signatures of a majority of the alumni and instructors of the college, were presented.

IMPORTANT ANNOUNCEMENT REGARDING ANNUAL MEETING

The Committee on Meetings announce that the Society has reached the point where in the future it will be necessary for papers to be

submitted earlier than has heretofore been the custom, in order to secure consideration for any given meeting. In order to avoid disappointment on the part of those expecting to contribute, papers for the next Annual Meeting in December should be in the hands of the Secretary of the Society by September 1. The responses to the efforts of the Committee in securing papers of high engineering merit have been generous, and it is felt that the greatest good to all will be realized by allowing ample time for the advance publication of papers and the solicitation of discussion. The committee is making this early announcement so that there will be ample time to prepare manuscript by the date mentioned.

THE PRESENT STATE OF DEVELOPMENT OF LARGE STEAM TURBINES

By A. G. Christie

ABSTRACT OF PAPER

This paper deals with the present state of development of leading types of large steam turbines, some details of construction, the commercial results obtained and some new uses to which steam turbines have been put. It also points out the probable tendencies of steam-turbine development.

For the purposes of discussion, large steam turbines are divided into two types, fundamental and modified or combined. The weak elements of the fundamental types are discussed and the advantages of the new types pointed out. Some new constructons are shown in section.

The details of construction are discussed fully. A table with brief notes on the details of construction of individual types is included for rapid comparison of the practice of various manufacturers. The different types are also compared on the basis of efficiency as shown by published results of tests.

The present status of low-pressure turbines, turbo-compressors, turbo-driven pumps, geared turbines and marine turbines is discussed briefly. The closing paragraphs deal with the probable trend of steam turbine development and future possibilities.

THE PRESENT STATE OF DEVELOPMENT OF LARGE STEAM TURBINES

By A. G. Christie, Madison, Wis.

Associate Member of the Society

The Transactions of the Society for 1910 includes two papers on steam turbines, which, with the accompanying discussions, give interesting information on the present development of two prominent American types. Mr. Geo. A. Orrok discussed small steam turbines in his paper published in Vol. 31 of the Transactions. Vol. 25 contains papers and discussions presented at the Chicago meeting of the Society in 1904, which cover the leading types of steam turbines then in commercial use. It seems fitting now to record the present state of development of large steam turbine construction and of the field of the steam turbine both in this country and in Europe.

PRESENT FIELD OF LARGE STEAM TURBINES

2 Steam turbines are now being used for driving alternating-current generators, turbo-compressors, turbo-blowers, pumps and marine propellers, and, by means of gearing, to furnish power to direct-current generators, rolling mills and the propeller shafts of steamships. Reciprocating engines were formerly used for such purposes, but recently this class of engine has seldom been installed except for rolling-mill work, non-condensing service as on heating systems, rope and belt drives, hoists, and in certain combinations with low-pressure turbines in marine work. The high economy of the piston-pumping engines and also of some types of air compressors, has continued their popularity in spite of the increasing competition of steam turbine units. The steam turbine has found favor principally on account of its low first cost of installation, its small

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

floor space requirements, its continued good steam economy over a period of years and its small operating and repair charges.

3 The increased use of steam turbines in sizes up to 1000 h. p. seems to have received at least a temporary check in Europe by the introduction of the new Stumpf direct-flow engine.

TYPES OF STEAM TURBINES

4 For the purpose of this paper, large commerical steam turbines will be divided into two classes: (*a*) fundamental types, and (*b*) modified or combined types.

5 The fundamental types of turbines are as follows:

a The Parsons type, which works on the so-called "reaction" principle. In this type the heat energy of the steam is changed into kinetic energy, both in the stationary guide blades and in the moving blades. In other words, both sets of blades act as orifices expanding the steam through a small pressure drop. As nozzles and orifices usually have very high efficiencies, this turbine should, theoretically, prove the most economical of all types. The construction of the Parsons turbine is familiar to all engineers. It consists of a drum, or a number of drums, carrying the blade rows which alternate with rows in the casing. The drums carry balance pistons to equalize the end thrust.

b The Curtis type, which works on the impulse principle with high steam velocities and few stages. Each stage, however, is provided with two or more rows of revolving blades known as "velocity rows," with intermediate rows of guide blades. The steam velocity at the beginning of each stage is high. The characteristics of this type of turbine are presented in Mr. W. L. R. Emmet's paper, The Steam Turbine in Modern Engineering.[1] The revolving blades are carried on disks separated by diaphragms, which extend to the shaft and which carry the orifices between stages. Curtis turbines are now usually built with horizontal shafts. In American practice some sizes over 7000 kw. still have vertical shafts.

c The Rateau turbine, which consists of a number of simple impulse wheels in series on the same shaft and separated from one another by diaphragms carrying nozzles. It

[1] Trans. Am. Soc. M. E., vol. 25, p. 1041.

operates with lower steam velocities than the Curtis and consequently has many more stages. Each revolving element carries only one row of blades. The characteristics of this type will be found in Prof. A. Rateau's paper, Different Applications of Steam Turbines.[1]

The type of turbine known as the Zoelly belongs to the same classification as the Rateau, from which it differs only in the use of higher steam velocities, in the number of stages and in certain constructional details. Fig. 1 shows a section of a Zoelly turbine. Though this type has been widely adopted abroad, it has not been placed on the American market up to the present time.

6 Each of these fundamental types is based on sound theoretical principles. In the process of manufacture and in operation, certain features have not proved entirely satisfactory, hence far-reaching modifications have been made in the design of some types of turbines. Some manufacturers have combined the characteristics of two or more types to overcome the inherent limitations of each fundamental type. A discussion of the unsatisfactory operating conditions of each type will show the reasons which led up to the changes in recent turbines and will also aid in distinguishing the novel features of new designs.

7 The first rows of spindle blades in a standard Parsons turbine are placed on a drum of small diameter in order to make the blades as long as possible and to minimize the proportional leakage losses past the ends of the blades. A large number of rows are provided in order to keep the steam velocities low, as the blade velocities must be low with the small drum. The drop in pressure at each row is small and hence the leakage is correspondingly reduced. This construction results in a turbine with a long spindle and with great distance between bearings. High-pressure steam, frequently at high temperature, is admitted to the casing. Distortion of casing and spindle are thus easily conceivable in such construction, and to allow for this contingency the clearance on the ends of all blades is usually increased. This distortion may be due either to unequal heating or to the growth of the cast-iron casings. The fluid friction losses are large in this high-pressure section, for a large number of rows of blades must be revolved in steam of high density. The leakage losses and fluid friction losses in the high-pressure section, to-

[1] Trans. Am. Soc. M. E., vol. 25, p. 782.

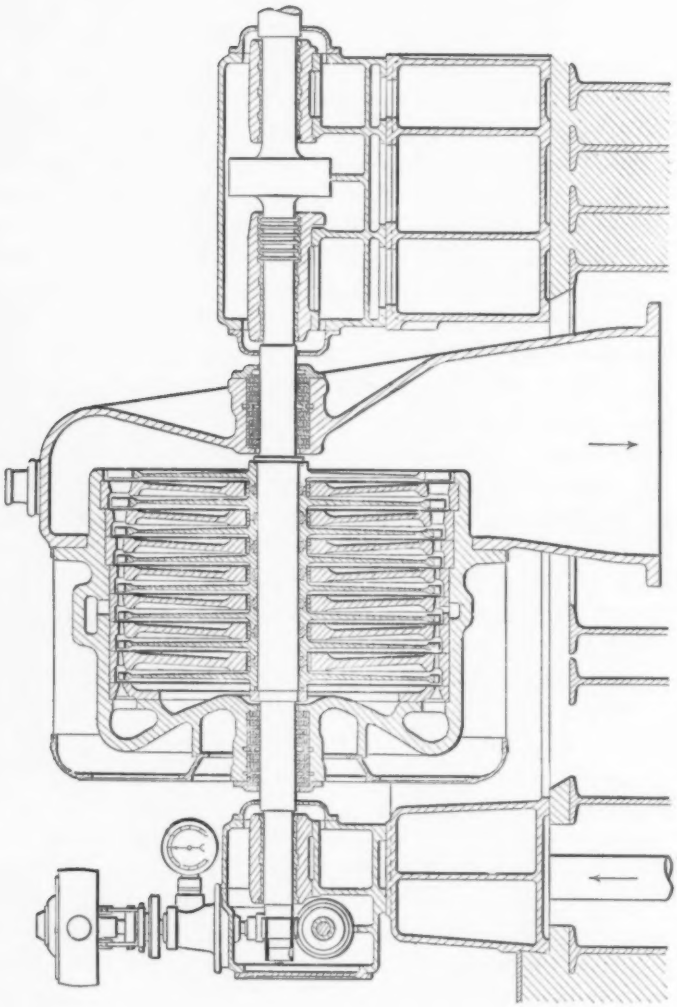

FIG. 1 SECTION OF ZOELLY TURBINE

gether with the troubles due to distortion in the long shaft, have forced designers to introduce modifications in this portion of the turbine, either by new blading arrangements, by dividing the total expansion in large sizes between two cylinders or by the introduction of impulse blading.

8 The low-pressure sections of Parsons turbines have always shown high efficiencies. As low steam velocities are characteristic of this type, there is no cutting away of blade material, even with very wet steam, provided no injurious properties are present in the feedwater. This low-pressure section has therefore been altered only in details.

9 The presence of a large low-pressure balance piston in close proximity to the steam inlet has frequently been the cause of serious distortions. Many builders now place this balance piston in the exhaust end, a construction due to Fullager.

10 The Curtis turbine utilizes high steam velocities in all stages. As steam becomes wet through expansion in the low-pressure stages, there is frequently considerable cutting of the blade material by the steam, although, contrary to first expectations, there is seldom cutting in the high-pressure blades due to the high initial velocities. The first row of velocity blades in a stage usually does the greater portion of the work, and hence the second row does not work at maximum efficiency. As the steam of the first stage is expanded very fully in the nozzles, there is no high pressure or superheat in the turbine casing or at the glands. The vertical type of unit is sometimes subject to electrical unbalancing and to other troubles peculiar to this construction. It is not as accessible in operation as the horizontal machines.

11 Recent designs have provided for horizontal units and for the replacement of the low-pressure section by sections of other types.

12 The Rateau turbine has high pressures on the gland at one end. There are a large number of disks revolving in dense steam at the high-pressure end. It has lower steam velocities throughout than the Curtis, and consequently has no blade-cutting effects in the low-pressure section. The clearances around the blades are large, but the shaft clearances of the diaphragms must be kept small. Some builders of this and the Curtis type have brought out new designs which employ the high-pressure section of the Curtis with the low-pressure section of the Rateau. These represent a compromise between efficiency and manufacturing costs.

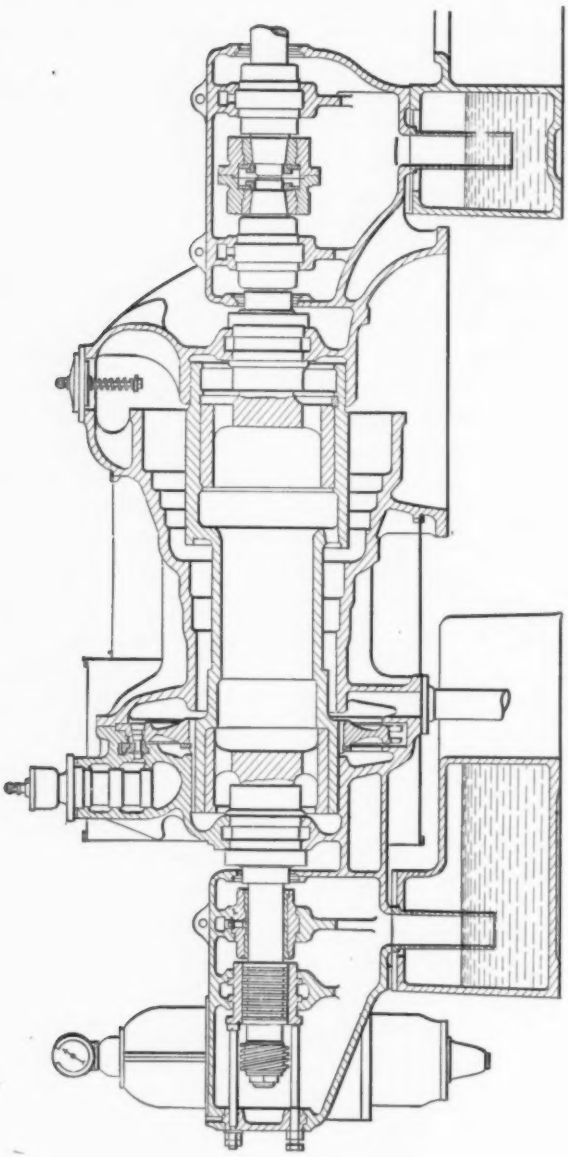

FIG. 2 SECTION OF BROWN BOVERI'S CURTIS-PARSONS TURBINE

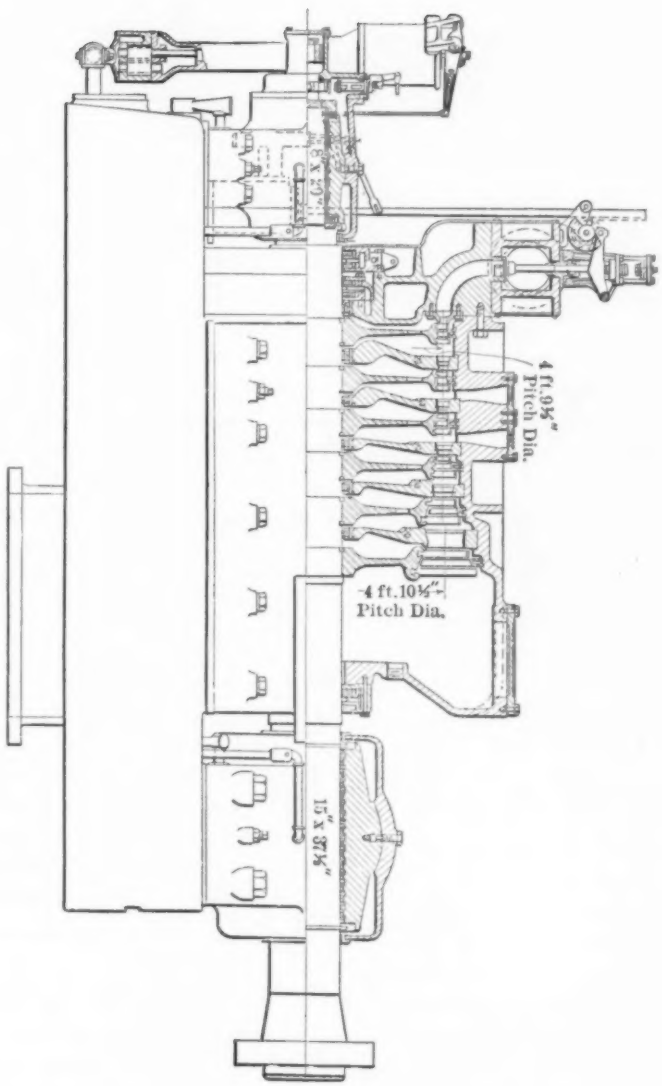

FIG. 3 HORIZONTAL CURTIS STEAM TURBINE, 7000 KW., 1800 R.P.M., SIX-STAGE

13 Under the classification of modified or combined types, there are turbines with modified Parsons, Curtis and Rateau construction, and turbines with combinations of Curtis-Parsons, Rateau-Parsons, Curtis-Rateau and Curtis-Zoelly construction, with a few special combinations. Some typical turbines of these classes are described in the following paragraphs.

14 The Allis-Chalmers Company manufacture a turbine of the modified Parsons type. High peripheral speeds are employed with a decreased number of rows. A portion of the theoretical efficiency in the high-pressure end has been sacrificed by the use of fewer blade rows. Also a smaller proportion of work is done in this section than is usual in Parsons turbines. The blades are all provided with a channel-shaped shroud. European experimenters have pointed out that better efficiencies are obtained with shrouded than with plain blades, as the so-called spilling-over at the ends is prevented. Wing blades are fitted in the last low-pressure rows to take care of the large volumes of low-pressure steam. The spindle is much shorter and stiffer than the standard Parsons, and hence smaller clearances can be provided. The Fullager low-pressure balance piston is also used. The outstanding features of this design are, reaction principle with drum construction, few rows of blades with high steam velocities, short spindle construction, and employment of wing blades for high vacuum. The steam consumptions obtained on this type show improved efficiency over standard construction.

15 A number of Parsons turbines have been built in which the total expansion is divided between two cylinders. One of the best known units of this type is installed at Dunstan Power Station, England. This was built by Richardsons Westgarth & Company on Brown-Boveri designs. This unit, however, employs a modified double-flow Parsons construction in the low-pressure cylinder.

16 C. A. Parsons & Co. have used cast-steel cylinders for the high-pressure portion of their turbines so as to overcome the troubles due to deformation and growth of metal under high pressures and superheat experienced with cast-iron casings.

17 The Westinghouse Machine Company have developed a double-flow machine which employs a Curtis high-pressure stage with Parsons intermediate and double-flow low-pressure sections. There is only one balance piston in this machine next to the Curtis ring. Mr. Sam L. Naphtaly described this type of machine in his

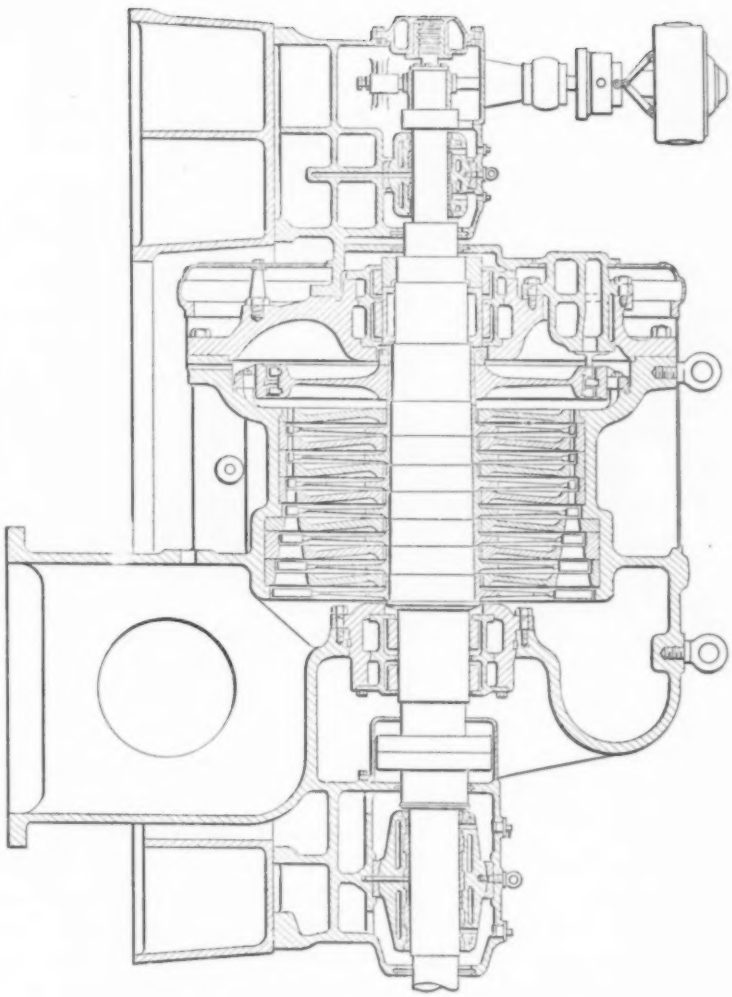

FIG. 4 SECTION OF BERGMANN CURTIS-RATEAU TURBINE

paper, "Test of a 10,000-Kw. Steam Turbine," and also showed a cross-section of the turbine.[1]

18 These turbines run at high speeds, have short shafts and small clearances. The steam is expanded in the nozzles, and hence there are no high pressures or temperatures in the casing. By providing two low-pressure sections, high vacuum can be economically utilized.

19 European builders of Parsons turbines, among whom are Brown Boveri & Company, C. A. Parsons & Co., Franco Tosi, Sulzers, Willans & Robinson and Erste Brünner, have replaced the high-pressure sections of their Parsons turbines by a Curtis wheel with two or more velocity rows, but have retained the single-flow Parsons drum construction for the remaining portions. One of these units is shown in Fig. 2. The temperatures and pressures in the casing are low as the steam is expanded in the nozzles. The distance between bearings is decreased, the shaft is stiffer and clearances are smaller than in the standard Parsons turbine. The Fullager balance piston is used in the turbines of several of these builders. Turbines built in this manner have shown some exceptionally good efficiencies.

20 Compared with the Westinghouse machine, the leakage at the end of the low-pressure blading is less than in the double-flow section. On the other hand, the Westinghouse machine has smaller balance piston losses and can also utilize the highest vacuum at better efficiency.

21 Melms & Pfenniger employ a drum impulse section of about five stages instead of a Curtis ring on some of their large turbines. They claim improved efficiency from this construction, though high-pressure and high-temperature steam are introduced into the casing.

22 The General Electric Company now manufacture a horizontal type Curtis turbine in all sizes up to 7000 kw., a section of which is shown in Fig. 3. This embodies all the essential features of the Curtis design. Compared with the vertical type, this design provides easier access to all working parts such as governor, bearings, valves, etc., and allows a better survey of the unit. The machine can also be dismantled and its internal parts examined with less trouble. The oiling problem is very simple compared with the vertical units, as there is no step bearing to provide for. It is possible that only horizontal units of all sizes will be built in the near

[1] Trans. Am. Soc. M. E., vol. 32, p. 1251.

future. The A. E. G. also builds similar turbines up to 1000 kw. capacity.

23 Many manufacturers in Europe are now building a turbine of the type shown in Fig. 4. This consists of a Curtis high-pressure with Rateau or Zoelly low-pressure sections, the number of stages depending on the size of the machine, the speed and the steam conditions. The pressure and temperature in the casing are low, as a consequence of the expansion of the steam in the nozzles. The steam velocity is moderate in the low-pressure section. This ma-

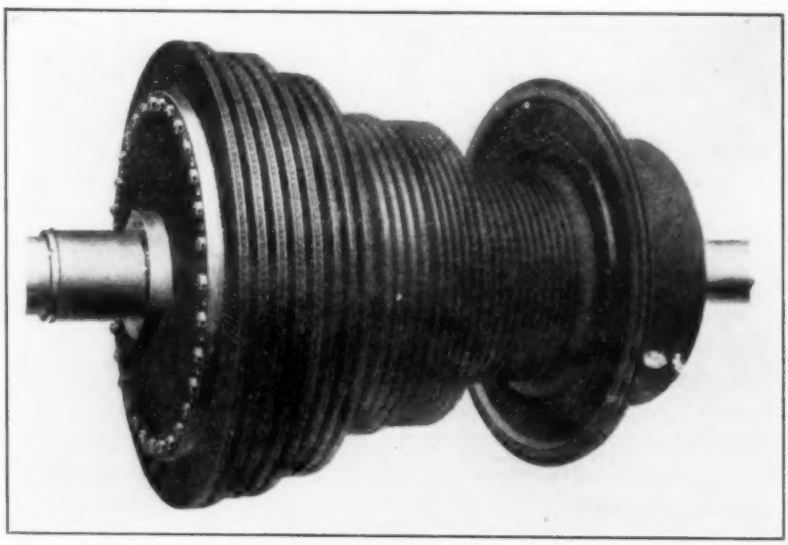

FIG. 5 SPINDLE OF CURTIS DRUM-IMPULSE TURBINE OF BELLISS & MORCOM

chine is longer than the simple Curtis but shorter than the Rateau. Its builders claim improved efficiencies over either of the fundamental types.

24 Belliss & Morcom, of Birmingham, England, have introduced a turbine, the spindle of which is shown in Fig. 5. It consists of a high-pressure Curtis stage with a low-pressure drum construction like Parsons, but fitted with impulse blading on the spindle and blades forming expansion parallel wall nozzles in the casing. All other impulse turbines employ diaphragms extending to the shaft. There are a greater number of rows in a Belliss turbine than in a corresponding Rateau turbine, and hence a lesser pressure drop through each set of guide blades. The net leakage loss may thus be

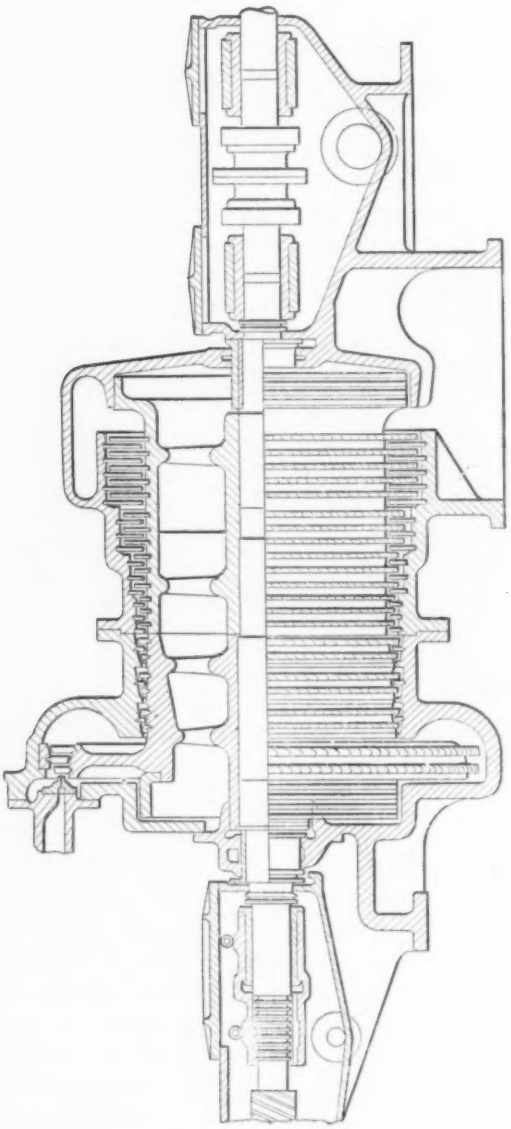

FIG. 6 SECTION OF CURTIS DRUM-IMPULSE PARSONS TURBINE

lower than the Rateau, even though the leakage area is greater. This turbine is essentially a Curtis-Parsons construction, with few rows of blades and with the expansion taking place in the stationary blades only.

25 A turbine, shown in Fig. 6, has recently been patented in England, which has some features of unusual interest. The high-pressure stage is Curtis, the intermediate section is of a drum impulse type with three stages, while the low-pressure portion is Parsons, which may be also double-flow. This combines features of all three fundamental types, and it will be interesting to see what results are obtained in practice by such an arrangement.

DISCUSSION OF TURBINE DETAILS

26 In the early days of steam turbine building it was difficult to secure suitable materials to withstand the stresses set up at high speeds of rotation. But as the demand for such materials increased, much study was given to the requirements for this service so that it has been possible through the use of more suitable material to increase very considerably the speeds of all sizes of steam turbines.

27 Higher steam velocities are possible with increased peripheral speed, and thus fewer rows of blades or stages are required. This results in a shorter and more rigid shaft construction, which is therefore less liable to vibration. Many builders, especially those of Parsons turbines, have found this construction to give an increase in steam economy over the slow-speed types, so that the more compact modern high-speed machine is more desirable than the older type.

BLADING

28 The requirements of a satisfactory blading material are that it shall withstand without deformation stresses due to centrifugal force, and temperature or pressure changes, it shall not cut out with high velocities of steam, and that it shall withstand the corrosive effects of moisture. Parsons turbines have used special bronzes and copper-nickel alloys. Steel blades have also been used in some cases. In these turbines the steam velocities are low and there is usually no cutting on this account. The principal problem with this blading is to manufacture it cheaply and secure it in such a manner that it will withstand all stresses to which it is subjected. This blading design is therefore a question of detail, and, as shown in Table 1, many ingenious schemes have been devised.

29 In Parsons turbines the blades are usually cut, punched or pressed into proper form from strips of drawn material. The original Parsons blading consisted of alternate blades and distance pieces placed in a slightly dovetailed slot and calked tight. Many European builders thread blades and distance pieces on holding wires before calking in. In this case the blading is made up in sections. Other builders of turbines of the Parsons type use Sankey's solid foundation ring held in place by a soft metal calking strip. Allis-Chalmers employ this well-known construction. As this form is usually all machine made, it is considered by many engineers to be safer than where dependence is placed on hand work, such as must ordinarily be done where each distance piece is calked separately.

30 There are many methods in vogue for spacing and reinforcing the ends of Parsons blading. The Westinghouse Machine Company use a comma-shaped wire threaded through the blades near their outer end and bent over between them. Similar schemes are used by manufacturers in Europe. As a rule, however, European builders follow the old Parsons method of silver soldering or brazing the blades to a holding wire near their outer ends. They generally thin down the tips of the blades to reduce weight and to avoid injurious effects to spindle or casing from accidental rubs. Some builders, such as Sulzers, do not thin off their blades or use shrouds, but make their spindles so rigid and well balanced that the blades can be made with a very heavy cross-section and hence need no support. Several manufacturers rivet the outer ends of their blades into channel-shaped shroud rings. This gives an especially stiff construction.

31 Advocates of the shrouded blading claim that it provides a labyrinth passage for the steam and thus reduces the leakage losses from row to row. It also holds the blades at the required angles. It has been noted, however, that with wet steam there is a tendency for the moisture to pit the casing opposite the edges of the shrouds and thus increase the clearances. This action has also been noticed with unshrouded Parsons blading. The shrouded blading is usually so stiff that serious damage is done if rubbing starts between the blading and the spindle or casing.

32 The blading of impulse turbines is of nickel steel, frequently with 25 per cent nickel, in the high-pressure section and special bronze in the other stages. Experience with this 25 per cent nickel steel blading material has not been entirely satisfactory, and several manufacturers are now using a low carbon steel alloy with just sufficient nickel to prevent corrosion, usually about 5 per cent. These

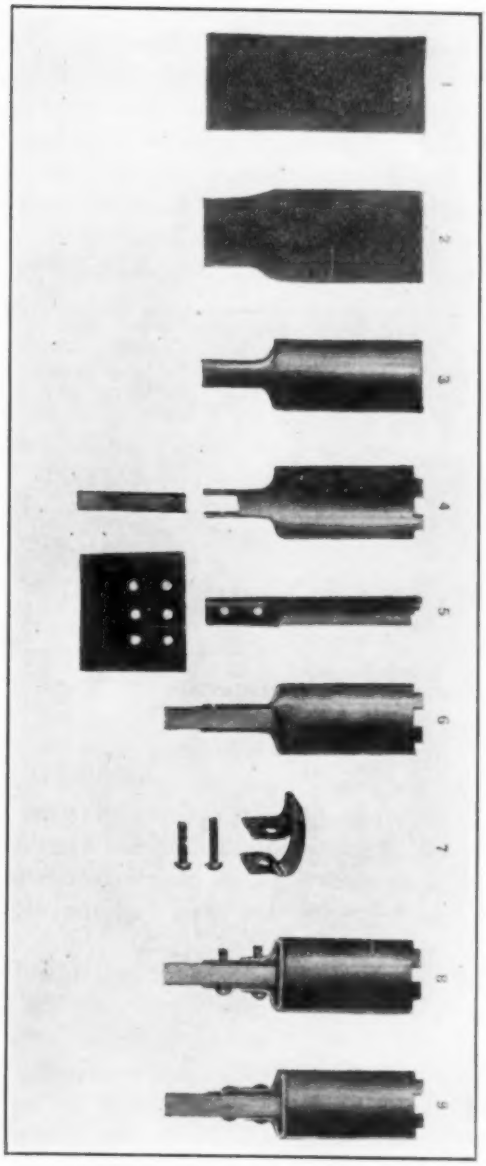

FIG. 7 METHOD OF MANUFACTURE OF BLADING FOR RATEAU SECTIONS OF BERGMANN TURBINE

blades are said to be stronger and less liable to fatigue of material. Special bronze and monel metal have also been successfully used. These impulse blades are stamped from sheets, drop forged or milled from solid bars with or without a wide base to act as a distance piece, or are made from extruded metal strips of the desired cross-section. Usually these blades are of crescent section, but some are formed of flat strip material and made of constant thickness over the width of the blade. The separate distance pieces are usually of the same material as the blades themselves. In general, all impulse blades are provided with shrouds to prevent vibration and also to provide an enclosed passageway for the steam at high velocities. As there

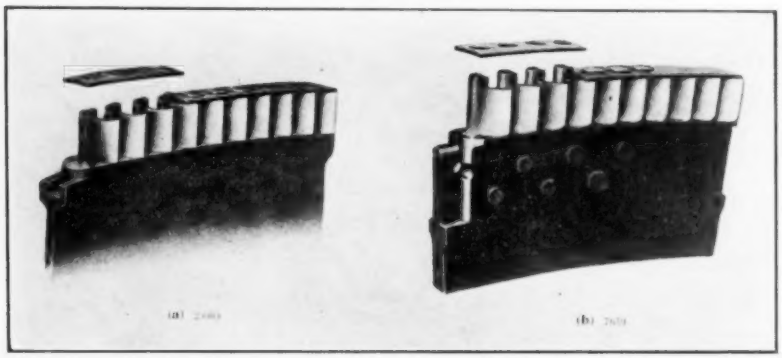

FIG. 8 TYPES OF BLADING USED BY BRITISH WESTINGHOUSE COMPANY

is no drop in pressure between the two sides of a row of moving blades, the clearance can be made large, both on the end and sides, so that there is little possibility of rubbing when in operation.

33 Impulse blading is usually held in place in dovetailed grooves or in tee-shaped slots, although some manufacturers form their blades with two legs which straddle the disks and are held firmly in place by rivets.

34 The first impulse turbines had blading in which the inlet and discharge angles were equal. Now almost all builders use blading on which the discharge side of the blade makes a sharper angle with the axis than the inlet side. This does not necessarily mean that the discharge area of the blades is smaller than the inlet area, for the blade is usually lengthened radially on the discharge side. Thus both inlet and discharge areas are made equal. The sharper angle of discharge reduces slightly the relative velocity of exhaust from the moving blade. On turbines of the Curtis type there is usually

only a small difference between the inlet and outlet angles of the first row of moving blades, but on the second row of moving blades in the stage, the entrance and exit angles often differ by as much as 15 deg.

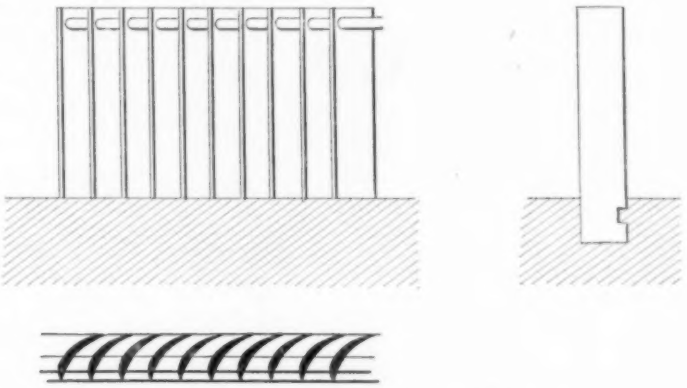

FIG. 9 BROWN BOVERI'S REACTION BLADING

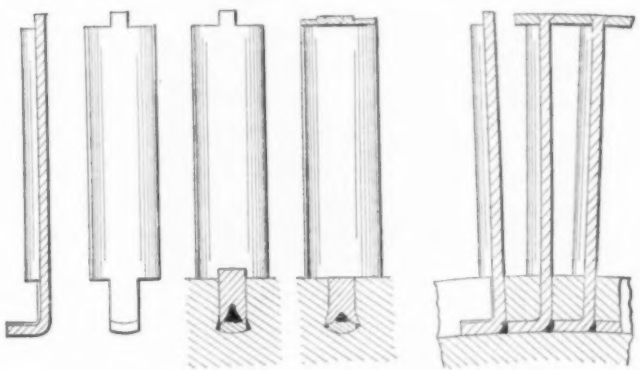

FIG. 10 BELLISS & MORCOM IMPULSE BLADING

35 Some impulse turbine designs are such that there must evidently be some such reaction effect in the moving blades as is obtained in Parsons turbines, though not of sufficient amount to cause any noticeable end thrust.

36 Some recent interesting developments in blading are shown in Figs. 7-11. Fig. 7 shows the various steps in the manufacture of the thin sheet metal blading used by Bergmann. The blading of the Rateau disks of British Westinghouse turbines is clearly illustrated in Fig. 8. Brown Boveri's form of Parsons blading is illustrated in

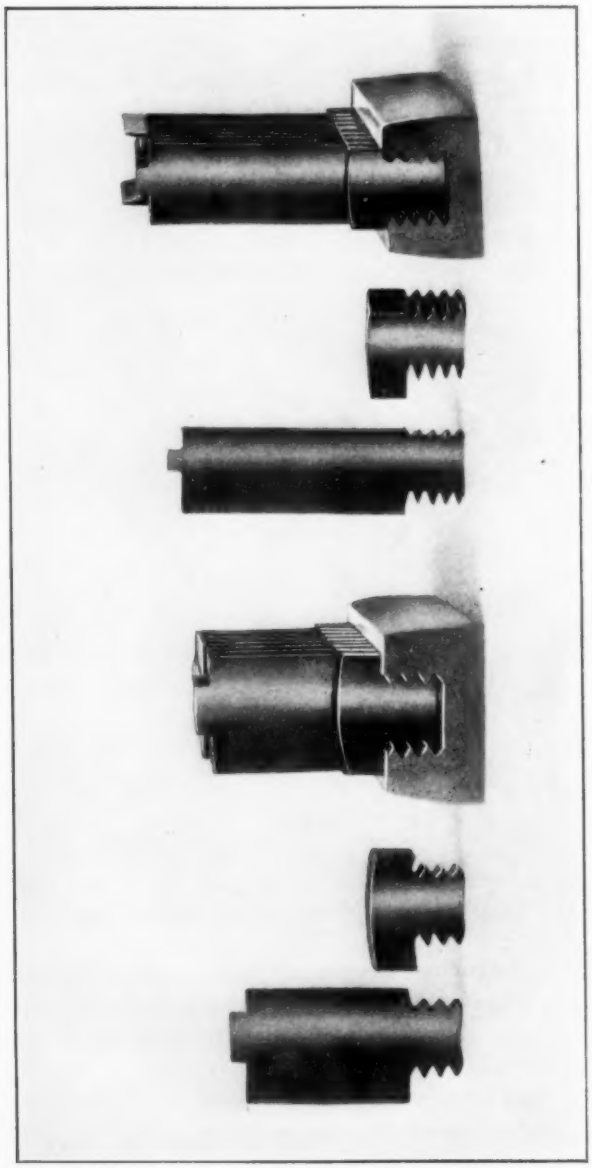

FIG. 11 FRANCO TOSI'S BLADING. IMPULSE BLADES ON LEFT, REACTION BLADES ON RIGHT

Fig. 9. Belliss & Morcom use a unique form of blading on the drum impulse section. This blading is shown in Fig. 10. Franco Tosi cuts projections like threads in his blade grooves and mills corresponding projections on the base of the blades, as in Fig. 11. This is a very satisfactory but expensive form of blading.

37 The number of rows of blades in any given type depends entirely on the size and speed of the unit, the steam conditions under which the turbine will operate and the heat drop per row or stage assumed in the design. Each designer selects such conditions as his experience and judgment lead him to consider the most satisfactory. It is thus hard to draw any general conclusions as to the most desirable number of rows for any class of turbine. In general, it may be said that the Parsons turbines require from 40 to 80 sets of moving and stationary blades, the Curtis from 4 to 8 stages, the Rateau from 12 to 25 stages, the Zoelly from 8 to 20 stages, while the Curtis-Parsons and Curtis-Rateau combinations of course require several rows less than the fundamental types.

NOZZLES

38 The high-pressure nozzles in impulse or combination turbines are made of bronze or nickel steel highly polished inside and placed either in the upper end of the casing itself, or, more preferably, in a separate steel casting which bolts upon the casing. In the latter construction the casing itself is never subjected to the high temperature or pressure of the entering steam.

39 The nozzles in the diaphragms between stages are usually made of nickel steel or other special steel, bent to the proper form and cast in place in the diaphragm body. Some manufacturers use brass nozzles in place of steel made up in sections and riveted or bolted in place. All these passages must be smooth and preferably very highly polished to reduce friction losses. After the first stage all nozzles have parallel walls on the discharge side.

BEARINGS AND LUBRICATION

40 Practice varies widely with regard to the design and construction of journals and bearings. Builders of impulse turbines invariably use cast-iron bearing shells provided with spherical self-aligning pads and lined with white metal. These are being used to an increasing extent on Parsons turbines and without exception on all low-speed units. Some manufacturers still retain the original Parsons form of bronze bearing shell with concentric rings on the

outside, separated from each other by oil films. It has been claimed for this type of bearing that the oil films dampen any slight variation of the spindle and thus provide a quieter running machine. Experience has shown that this is not always the case and that such a bearing is often a real source of danger when the added clearance of the spindle due to play between the rings is taken into consideration. This construction is also much more expensive than the white metal bearings.

41 In Europe many turbines are run with a minimum amount of oil and with oil leaving the bearings at a temperature of 190 deg. fahr. This practice is based on the argument that such a system of lubrication requires the least expenditure of power for oil circulation and in friction losses. However, practice seems to be tending towards flooded lubrication, in which a great quantity of oil at a temperature of about 100 deg. fahr. is forced through the bearings by a pump of the rotary, centrifugal or gear type, driven from the main shaft of the turbine. The oil pressure at the bearings varies from 3 to 20 lb. per sq. in. The life of the oil is much longer in this system than with very hot oil and any wear on the bearings themselves is absolutely prevented. Occasionally the bearing shells are water-cooled, but this practice should be discouraged. Cooling water can be used much more satisfactorily outside of small oil pipes in properly designed coolers. In case the water is dirty or full of scale-forming impurities, these can be more readily removed from an oil cooler than from the interior of a bearing shell.

42 Flooded lubrication has enabled manufacturers to cut down the length of their bearings and thus reduce the total length of their turbines. The increased pressures per unit area on the bearings have not introduced any difficulties, so that pressures of 80 to 100 lb. per sq. in. at a surface speed of 60 ft. per minute are common practice. The best results are usually obtained with a temperature of about 125 deg. fahr. as the oil leaves the bearings.

SPINDLE CONSTRUCTION

43 Parsons turbines in America are usually built with a hollow quill into which the journal ends are forced and fastened by shrink links or bolts. The high-pressure blading is placed in grooves in one end of the quill itself. The intermediate and low-pressure blades are usually carried on cast or forged-steel rings which are afterwards forced and keyed upon the central quill. In Europe excellent hollow steel forgings can be obtained very readily, and hence the spindles

of Parsons and other drum type turbines are usually made up of one forging with the journal shafts fastened into the ends.

44 The shafts of impulse turbines are usually in one piece and carry the blade disks, which are high-grade steel or nickel-steel forgings or castings. These disks are fitted and keyed on the shaft and held in place by shrink links or lock nuts.

45 It is quite general practice now to design the shafts so that the normal speed of the turbine will be very considerably below the critical speed due to any slight unbalancing of the mass that may be present. This removes the dangerous vibrations often experienced when passing through critical speeds and permits closer clearances to be used on Parsons blading and in the labyrinth passages in the diaphragms between the stages of impulse turbines.

PACKING GLANDS

46 A small impeller supplied with water is provided on all American built Parsons turbines to form an air seal at the shaft glands. European builders prefer to use labyrinth packings with live or throttled steam as an air seal. The objection raised by foreign builders to the water packing is that it takes too much power to drive the impeller, and that it provides a condensing surface for the steam. The amount of water required in a well-designed gland is very small, and there is no great circulation. Thus the water can have only a small effect as a condensing medium. Usually the steam directly inside the casing is under vacuum and then condensation would not be objectionable. It requires considerable steam to pack the labyrinth type of gland, and this loss often exceeds that due to the power required to drive the water impellers.

47 Impulse turbines use carbon rings at the high-pressure gland, especially when superheated steam is used. The leakage past the first rings is carried through a passage and pipe to the low-pressure glands to act as a vacuum seal. The labyrinth packing in the diaphragms is usually of bronze in the high-pressure stages and frequently of white metal in the low-pressure section. The low-pressure shaft glands are made either with carbon or labyrinth packing, sealed by steam.

THRUST BEARINGS

48 Thrust bearings are now provided on all types of turbines. These serve to adjust the position of the spindle and to take up any end thrust present. The end thrust in well designed turbines is

usually of small amount and seldom causes trouble. These bearings are usually flooded with oil and are often made with bronze rings which can easily be replaced in case of damage.

49 Several builders of Curtis-Parsons turbines have substituted an oil thrust piston for the steam balance piston of the usual form. This piston is placed on the governor end of the shaft outside of the casing proper. The thrust is taken up by oil which is supplied under pressure and which can escape only through sets of labyrinth baffles on the piston and on the surrounding chamber. The piston adjusts itself for the amount of thrust present by a small axial movement of the shaft, thus opening or closing the discharge area for the oil through the labyrinth baffles, and automatically varies the oil pressure on the face of the piston. This arrangement does away with the loss of steam through balance pistons and should improve the turbine efficiency, though a small amount of power is required to pump the oil.

GOVERNING DEVICES

50 The speed of Parsons turbines is usually controlled by a centrifugal governor of the Hartung or similar type which regulates the position of a balanced poppet valve through the medium of a steam or oil relay. The latter type is coming into more general use on account of its many advantages over the steam relay. The steam is throttled either at constant pressure or by a pulsating action.

51 Impulse turbines and turbines employing a Curtis high-pressure ring usually govern by means of Hartung type governors and oil relays. The speed is sometimes controlled by simple throttling of the steam, which practice is common in Europe. In this case, additional nozzles can be opened or closed by hand as required. This system would be unsuitable with violently fluctuating loads. Other types employ both throttling and automatic nozzle regulation, while, again, many turbines, particularly of the Curtis type, are built for nozzle governing alone. In Europe there is a difference of opinion as to the most economical method of governing. M. H. Zoelly claims that he gets the best results on his type of turbine by throttle governing. It is generally admitted, however, that impulse turbines give the best results by controlling the number of nozzles that are open at any load. With such control the pressure before the nozzles in service is always the normal pressure for which these are designed. The General Electric Company govern their turbines above 300 kw. by means of an oil relay system which oper-

ates through a piston and camshaft to open or close nozzles as required by load conditions.

52 The Westinghouse Machine Company use a vibrating oil relay system on many of their machines. Recently they have adopted on some sizes a very powerful governor which is direct-connected to the governing valve and operates it without the use of relays. This system was adopted by some of the early European builders, but was abandoned in favor of the oil relay system.

53 Sulzers have recently installed turbines in which the conventional centrifugal force governor has been replaced by a hydraulic governor. This governor consists of a simple centrifugal pump, geared to the main shaft and delivering oil under pressure into a chamber beneath a spring loaded piston. The pressure under this piston is thus dependent on the speed of the main turbine and the piston's position in its cylinder will vary accordingly. The piston is connected to the usual balanced valve of an oil relay which controls the oil supply from the main oil pump to the throttle valve in the usual manner. There are no mechanical parts to wear in this arrangement. The apparatus is extremely simple and has many distinct advantages.

54 All turbines are now provided with a small overspeed governor, usually placed at the outside end of the shaft. This operates at a determined percentage over speed and closes the main or secondary steam valve either by means of a steam or oil relay or by a falling weight through a system of levers and springs. The oil relay system has the advantage that the valve shuts immediately should the oil supply fail for any reason.

55 Parsons turbines are usually provided with a secondary overload valve which automatically admits live steam to the second diameter of blades. Impulse turbines have additional sets of nozzles with valves which may be opened automatically, or by hand in case of overloads.

CASINGS

56 Practice varies widely in regard to the construction of turbine casings. These are generally made of cast iron, though some European builders make the high-pressure front end of cast steel. Parsons turbines are built with the top and bottom halves single castings or made in sections. Some of the older designs of impulse turbines used solid diaphragm plates placed on the shaft between the disks. The clumsiness of handling and the difficulty of making repairs with

this construction has forced most builders to make these in halves and to fasten them to the top and bottom portions of the casing.

57 Cylinder casings are now made of symmetrical design and without any deep metal webs or ribs as stiffeners on the outside. Equalizing pipes with provision for expansion take the place of passages formerly cast in the casings themselves. Strains due to unequal temperatures must be avoided in all portions.

BED-PLATES

58 Several European builders make a practice of filling the hollow portions of their turbine and generator bed-plates with concrete after erection. This adds more mass to the turbine unit and is said to dampen any slight vibration that may be present.

TURBO-GENERATORS

59 A discussion of the design of turbo-generators is beyond the limits of this paper. In general, European builders of electrical machinery allow less overload capacity than is usual on American machines, but give better guarantees of efficiency and regulation.

60 Practice varies widely in regard to the normal rating and maximum capacities of different turbines. Some European builders guarantee their generators to carry normal full load without undue heating only for two to six hours. Others follow the practice which has been introduced in America of rating their turbines at the maximum load they will carry continuously. Several builders still offer turbines which will carry as high as 25 per cent overload continuously.

61 Such conditions are very confusing to purchasers. The time seems opportune for this Society to fix a standard for the rating of steam turbines and to define the overload capacity which may be expected of this type of engine.

COMMERCIAL CONSIDERATIONS

62 When steam turbines are to be installed at high altitudes, as in many locations in the west, the effect of altitude on economy frequently comes up. It can be easily shown that with the same boiler pressure and the same absolute pressure in the condenser, the steam consumption of a turbine at 5000 ft. elevation should exceed that of a similar turbine at sea level by less than 1 per cent.

63 Each turbine is designed to operate at maximum efficiency at some given vacuum. Owing to uncertainties in design factors

for losses and to slight inaccuracies in construction, the most efficient vacuum may vary somewhat from that for which the turbine was designed. Theoretically, the steam consumption should decrease as the vacuum increases, but this rate of decrease will vary for each type of turbine, depending on the blade areas and steam velocities of the low-pressure section. It is, therefore, essential that this rate of change be determined for each individual turbine by actual test.

64 In the purchase of steam turbines, operating efficiency and costs should be considered together with first cost, and the machine selected should be the one on which the ultimate operating and fixed charges are a minimum. This practice is followed in Europe. In America the bargain-counter idea unfortunately possesses many engineers, and manufacturers have frequently been forced to sacrifice efficiency in order to meet competition.

65 In Table 1 some data are presented in tabulated form regarding various types of steam turbines. This information was gathered during personal visits to the various works and also by correspondence. Designs of steam turbine details are constantly changing, so that many of the items in this table may not correctly represent the latest practice of the various builders.

RESULTS OF TESTS

66 The efficiency of a steam turbine may be expressed in terms of pounds of steam per kw-hr., as an efficiency ratio or as the b.t.u. required per kw-hr.

67 The steam consumption is dependent on the initial steam pressure, its temperature or quality, and the condenser pressure. These factors vary in almost every test and the effect of a variation in each is not the same for all classes of turbines. Hence, different turbines cannot usually be compared satisfactorily on the basis of their steam consumption alone.

68 If the steam could expand freely to exhaust pressure in a turbine without radiation, friction, eddy or windage losses, its expansion would be adiabatic and on the Rankine cycle. The "efficiency ratio" expresses the proportion of the heat actually turned into work to that available from such an adiabatic expansion. In other words, it expresses the efficiency of the actual turbine as compared to the ideal turbine, and is independent of the type of turbine.

69 The b.t.u. per kw-hr. is figured above the heat of the liquid at exhaust pressure. This is not a satisfactory standard by which

TABLE 1 DATA ON STEAM TURBINE CONSTRUCTION

Manufacturer	Type of Turbine	R.P.M. of Various Sizes	No. of Stages.	Blade Speed in Ft. per Sec.	Steam Velocities, Ft. per Sec.	Blade Material
C. A. Parsons & Co.	Parsons & Curtis-Parsons	.....	Usually about 70	270 on low-pressure blades	540	Special bronze and delta metal.
Westinghouse Mach. Co.	Parsons & Curtis-Parsons	Up to 3000 kw., 3600; 3000-5000 kw., 1800; over 5000 kw. (750 or 720)	.....	Curtis blades 600; Parsons blades 350	Curtis 2000; Parsons 600	Special bronze and soft steel; Curtis blades bronze
Willans & Robinson.	Curtis-Parsons	.....	1 Curtis; 12-20 rows Parsons on 1 diameter of drum	.....	.....	Bronze and copper-nickel blades
Brown Boveri & Co., Richardsons Westgarth & Co.	Parsons & Curtis-Parsons	Up to 3000 kw., 3000; 3000-7500 kw., 1500	.....	Curtis blades 460 Parsons blades 380	.....	Curtis blades of special alloy; Parsons blades of bronze
Brush Electrical Engrg. Co.	Curtis-Parsons	.....	.....	.....	.....	Parsons blades bronze alloy
Erste Brünner M. F. G.	Curtis-Parsons	Up to 1500 kw., 3000; 1500-4000 kw., 1500; 4000 kw., up, 960	1 Curtis; remainder Parsons on 2 diameters of drums	.....	.....	Both Parsons and Curtis blades of special bronze
Franco Tosi.	Curtis-Parsons	Up to 2500 kw., 3000; 2500-8000 kw., 1500; 8000-12,000 kw., 1000	1 Curtis; remainder Parsons on 1 diameter of drum	.....	Curtis 2130; Parsons 230-460	Parsons and Curtis blades of special bronze
Gebrüder Sulzer.	Curtis-Parsons	.....	.....	Curtis blades 360; Parsons blades 310	.....	Curtis blades all of special steel; Parsons blades of special bronze
Melms & Pfenninger, Breitfeld, Danek & Co.	Rateau-Parsons & Curtis-Parsons	Up to 4000 kw., 3000; 4000-9000 kw., 1500	.....	Up to 575	Impulse section u; to 2200; Parson up to 900	Rateau blades of special bronze; Parsons blades also special bronze
Allis-Chalmers Co.	Parsons	Up to 2000 kw., 3600; 2000-5000 kw., 1800	.....	Up to 370	400-800	Copper-nickel alloy
Allegemeine Elektricitäts Gesellschaft.	Curtis-Rateau	Up to 3000 kw., 3000; 3000-9000 kw., 1500; 9000 kw. and over, 1000	1 Curtis; 9-12 Rateau	.....	.....	Bronze alloy or nickel-steel

Maschinenfabrik Augsburg Nurnberg.	Curtis-Zoelly	Up to 6000 kw., 3000; 6000-10,000 kw., 1500; 10,000-15,000 kw., 1000	1 Curtis; 6-8 Zoelly	Up to 650	Curtis 2500; Zoelly about 1300	Nickel-steel highly polished
British Westinghouse Co.	Curtis-Rateau	Up to 1500 kw., 3000; 1500-5000 kw., 1500	1 Curtis; 6-12 Rateau	300-500	Curtis 1500; Rateau 800	5% nickel-steel
Bergmann Electricitäts Werke.	Curtis-Rateau	.....	1 Curtis; 8-12 Zoelly	410-460	Curtis 2600; Rateau 1170	25% nickel-steel
Belliss & Morcom.	Curtis-Drum Impulse	Up to 1500 kw., 1800; use lower speeds than most builders	1 Curtis; 25-30 rows of impulse blades on drum	Not exceeding 400	.....	Curtis blades of steel; Impulse blades of phosphor bronze
Escher, Wyss & Co.	Zoelly	Up to 1500 kw., 3000; 1500-3500 kw., 1500; over 3500 kw., 1000	3000 r.p.m., 10 stages; 1500, 16 stages; 1000, 20 stages	.....	About 1400	5% nickel-steel
Maschinenfabrik Oerlikon.	Rateau	Up to 4000 kw., 3000; 4000-7500 kw., 1500; over 7500 kw., 750	10-16 stages with 2 or more mean diameters of blades	320-520	1310-1700	Nickel-steel
Sachsische Maschinenfabrik.	Zoelly	.....	5-6 on first diameter, 6-8 on second diameter	.....	.....	Nickel-steel
Frazer & Chalmers.	Rateau	.....	Are reducing number of stages on all turbines	Have increased blade speeds recently	.....	Nickel-steel
Pokorny & Wittekind.	Curtis-Rateau & Rateau	.....	.....	500-600	2300 admission	.....
General Electric Co.	Curtis	Up to 1400 kw., 3600; 1400-7000 kw., 1800; 7000-20,000 kw. (750 or 720)	4-7	460-500	About 1200 average	Special bronze extruded metal; Monel metal and steel used on some low-pressure blades
British Thomson Houston Co.	Curtis	Up to 1000 kw., 3600; 1000-5000 kw., 1800; up to 2000 kw., 3000; 2000-7500 kw., 1500	Same construction as General Electric Co.	600 as maximum	About 1800 from nozzles, 1100 from diaphragms	Special bronze and steel

TABLE 1—*Continued*

Manufacturer	Blade Form	Method of Blade Manufacture and Insertion
C. A. Parsons & Co.	Parsons form with very thick rounded inlet, blades thinned off at tips	Alternate distance pieces held by calking, ends of blades stiffened by silver soldering to a brass strip; Curtis blades shrouded
Westinghouse Mach. Co.	Blades crescent with heavy central portion and sharp inlet edge, dovetail base on impulse	Alternate distance pieces held by calking, ends stiffened by comma wire threaded through blades and bent over
Willans & Robinson.	Parsons rounded inlet, stamped to fit Sankey foundation ring and channel shroud; usual Curtis blades on impulse	Parsons sections made up by machine and held in place by calking in soft metal strip, final riveting of shroud completed last
Brown Boveri & Co., Richardsons Westgarth & Co.	Parsons blades not thinned off, usual form, base of blade notched to fit corresponding lateral projection in the grooves, blades not calked	Curtis blades in dovetail slot, no sharp corners on base and shrouded; Parsons blades threaded on wire and silver soldered
Brush Electrical Engrg. Co.	Usual Parsons form with calked distance pieces in dovetailed groove; also use grooved base of blades fitting corresponding grooves in shaft like teeth in a bolt	Tips of blades thinned off, blades silver soldered to wire strip
Erste Brünner M. F. G.	Curtis blades have special dovetail base and are separate from their distance pieces; Parsons blades usual form with distance pieces in dovetail groove	Curtis blades shrouded; Parsons blades thinned at tips and silver soldered to wire on side
Franco Tosi.	Blades and distance pieces milled at base, with thread-like projections to fit corresponding grooves in spindle	Curtis blades shrouded; Parsons blades also have channel shroud-riveted over after blades are in place
Gebrüder Sulzer.	Parsons blades of very heavy cross-section with standard Parsons distance piece and method of calking	Parsons blades not thinned at ends or reinforced in any way; Curtis blades milled to fit dovetail slot
Melms & Pfenninger, Breitfeld, Danek & Co.	Impulse blades have slot in side to fit projection in groove and held in place by soft metal calking strip; Parsons blades have special base to fit foundation ring and with special shroud on end	Parsons blade rings made up in halves and fitted to machine, sections held in place by soft metal calking strip, angle of blades, 20 to 45 deg.

Allis-Chalmers Co.	Parsons section with more defined inlet edge, blading uses Sankey foundation ring and shroud, blading made in sections outside turbine	Sections held in dovetailed slots by soft metal calking strips, shrouds channel-shaped
Allegemeine Electricitäts Gesellschaft.	Blade crescent shaped but of lighter section toward exhaust end, shrouded in groups	Blades cut from solid bar and set in dovetailed grooves with separate distance pieces
Maschinenfabrik Augsburg Nurnberg.	First Curtis rows have blades of heavy section, remaining rows are thin in section, blades are held in a ⊥ slot	Blades mostly drop-forged, all rows shrouded, outlet and inlet angles vary considerably
British Westinghouse Co.	Curtis blades have ⊥ base to fit corresponding groove; Rateau blades have 2 projections at base straddling disk and rivets rolled in place	Blade and distance pieces milled from solid steel bars
Bergmann Electricitäts Werke.	Blades stamped from sheet metal with projections at base straddling disk and riveted to it, blades of uniform thickness across section	Distance pieces drop-forged of special bronze and riveted with blades to disk, all blades shrouded
Belliss & Morcom.	Curtis blades, usual form; on impulse section blades are of uniform thickness over section and shrouded	Blades made with strip at base bent in form of L, distance piece formed with opening at base, the 2 projections spread over base of blade and fill dovetailed slot when calked down
Escher, Wyss & Co.	Blades stamped from sheet metal with ⊥ base, blades sometimes thinned off toward tips, distance piece stands well above surface of shaft	Blades milled with sharp edges and highly polished after stamping, distance pieces milled from solid bar, only thin shroud ring used
Maschinenfabrik Oerlikon.	Blades punched to shape from drawn metal strips, base is ⊥ shaped, simple band shroud ring used	Distance pieces milled, slot closed by special locking piece
Sachsische Maschinenfabrik.	Zoelly blades thinned off toward tips on low-pressure end, base ⊥ shaped with rounded corners	Blades and distance pieces milled from solid bar, channel-shaped shroud
Frazer & Chalmers.	Blades with forked base riveted to disks	Blades milled from solid bar, no shroud ring, but head milled on blades like De Laval
Pokorny & Wittekind.	High-pressure blades shrouded, low-pressure blades unshrouded on some units	.....
General Electric Co.	Blades milled with dovetail base and have shroud ring, distance pieces also milled	Blades put in place in special blading machine
British Thomson Houston Co.	Same construction as General Electric Co.	Same construction as General Electric Co.

TABLE 1—*Continued*

Manufacturer	Nozzles and Diaphragms	Shaft and Spindle Construction	Form and Material of Bearings
C. A. Parsons & Co.	.....	Spindles usually a single hollow forging, shaft ends fastened in	Parsons concentric bronze sleeve bearings in self-aligning cast-iron casings, large sizes white metal lined
Westinghouse Mach. Co.	.....	Shaft, hollow quill with blade rings forced on, ends held in by special patented fastening	Parsons concentric sleeve bearing on small units, cast-iron self-aligning bearings, white metal lined on large units
Willans & Robinson	.....	Disk, solid forging; drum, hollow rolled forging; ends fastened in by locked bolts; Fullager balance piston used	Cast-iron, white metal lined, self-aligning
Brown Boveri & Co., Richardsons Westgarth & Co.	Nozzles made in 3 parts of steel or bronze and all machined	Disk, solid forging forced and fastened on hollow forged drum spindle; shaft ends locked into drum; Fullager balance piston used	Parsons concentric bearings on high speeds, cast-iron, white metal lined, self-adjusting bearings on low speeds
Brush Electrical Engrg. Co.	.....	Disk, solid and fastened on solid or hollow forged drum spindle	Cast-iron, white metal lined, self-adjusting bearings on all sizes
Erste Brünner M. F. G.	Nozzles of special bronze alloy	Small sizes, nickel-steel, solid rotors; large sizes, hollow forged high grade steel; Curtis ring bolted to spindle	Parsons concentric sleeve bearings on small units; on large units cast-iron white metal lined self-aligning bearings
Franco Tosi.	Nozzles of steel or bronze, angles of nozzles 20 to 25 deg.	Forged steel shaft with spindle rings and impulse disk with balance piston forced on and held by bolts	Cast-iron, white metal lined, self-aligning bearings
Gebrüder Sulzer.	Nozzles of sheet brass cast in valve chest	Spindle, hollow forging with shaft ends and impulse disk bolted on	Cast-iron, white metal lined, self-aligning bearings
Melms & Pfenninger, Breitfeld, Danek & Co.	Nickel-steel guide blades in Rateau section with special labyrinth packing, angle of nozzles 18 deg. to 22 deg.	Spindle, hollow forging of nickel-steel, impulse disk and balance piston fastened on; shaft ends forced in	Parsons concentric sleeve bearings on small units, large units have cast-iron, white metal lined, self-aligning bearings
Allis-Chalmers Co.	.....	Hollow forged quill with blade rings of cast steel forced on; shaft ends forced into quill and held by bolts and buts; Fullager balance piston	Cast-iron, white metal lined, self-adjusting bearings

Allegemeine Electricitäts Gesellschaft.	Nozzles of bronze or nickel-steel, guide blades in diaphragms of nickel-steel cast in place	Solid shaft with disks forced on; disks of steel tapering toward the blades	Cast-iron, white metal lined bearings, usually water-cooled; only 3 bearings; very small governor bearing carried by front housing
Maschinenfabrik Augsburg, Nurnberg.	Nozzles of nickel-steel milled from solid, guide blades in diaphragms of highly polished sheet nickel-steel	Short stiff solid shaft running below critical speed disks of soft steel forced on shaft and locked; disks taper toward blade ends	Cast-iron, white metal lined, self-adjusting bearings; only 3 bearings used; governor bearing very small
British Westinghouse Co.	Nozzles of special steel, guide blades of special steel cast in place in diaphragms	Short, solid, stiff shaft stepped from center to receive disks which are of forged steel tapering from center to periphery	Cast-iron, white metal lined, self-adjusting bearings
Bergmann Electricitäts Werke.	Nickel-steel nozzles in 4 groups, nickel-steel guide blades cast in place in diaphragms	Short, solid shafts not too stiff, as shafts at 3000 r.p.m. and over, run above critical speed; disks forced on shaft and locked.	Cast-iron, white metal lined, self-adjusting water-cooled bearings
Belliss & Morcom.	Nozzles, steel or bronze, impulse guide blades of brass made in segments	Forged steel drum fastened to through solid shaft; drum carries impure disk and balance piston; heavy rigid construction.	Cast-iron, white metal lined, self-adjusting bearings
Escher, Wyss & Co.	All nozzles and guide blades of nickel-steel cast in place and highly polished	Solid shaft with disks supported on ⊥ ring so as to make shaft less rigid	Cast-iron, white metal lined, self-adjusting bearings
Maschinenfabrik Oerlikon.	Nickel-steel guide blades all cast in place	Nickel-steel shafts made heavy in the center; disks of forged soft steel forced on and locked to shaft	Cast-iron, white metal lined, self-aligning bearings, water-cooled in the larger sizes
Sachsische Maschinenfabrik.	Sheet-steel guide blades cast in place	Solid shaft with nickel-steel disks, highly polished, forced on	Cast-iron, white metal lined, self-adjusting, water-cooled bearings
Frazer & Chalmers.	Diaphragm guide blades cast in place	Solid shaft with disks forced on	Cast-iron, white metal lined, self-adjusting bearings
Pokorny & Wittekind.	Angle of nozzles 20 deg.	Solid shaft with disks forced on	Cast-iron, white metal lined bearings
General Electric Co.	Inlet nozzles of special bronze, steel guide blades in diaphragms cast in place and polished	Shaft, solid forgings; disks, cast steel forced on shaft	Cast-iron, white metal lined, self-adjusting bearings with copper cooling pipes cast in babbitt metal lining
British Thomson Houston Co.	Same construction as General Electric Co.	Shaft and disks of steel	.....

TABLE 1—*Continued*

Manufacturer	Surface Speed of Journals, Ft. per Sec.	Bearing Pressure, Lb. per Sq. In.	Type of Oil Pump	Governor and Method of Governing
C. A. Parsons & Co.	35-50	40-50	.....	Steam relay system, steam admitted in puffs, not continuous throttling
Westinghouse Mach. Co.	40-50	50-70	Rotary oil pump with sliding gate	Direct throttling from governor, sometimes pulsating; also pulsating oil relay
Willans & Robinson.	45-60	50-70	Both reciprocating and rotary types used	Direct throttling from governor
Brown Boveri & Co., Richardsons Westgarth & Co.	.....	.....	Rotary pump	Hartung type with oil relay, also slight pulsation to valve; valves on nozzles open automatically
Brush Electrical Engrg. Co.	.....	.....	Rotary pump	Steam relay system, pulsating action
Erste Brünner M. F. G.	50	40-50	Double rotary valveless pump	Hartung type governor, steam relay and piston valve on main inlet; pulsating governing
Franco Tosi.	.....	55-90	Rotary gear pump	Hartung type governor, oil relay system on main valve, automatic oil relay nozzle governing
Gebrüder Sulzer.	.....	.....	Triple gear pump	Automatic hydraulic regulator, centrifugal pump on oil only; throttling of supply; oil relay on all gear; no mechanical governor
Melms & Pfenninger, Breitfeld, Danek & Co.	40-80	35-85	Eccentric rotary pump	Hartung type oil relay, throttling governor
Allis-Chalmers Co.	50-60	60-140	Rotary lobe pump, auxiliary duplex pump for starting	Hartung type oil relay, throttling governor
Allegemeine Elektricitäts Gesellschaft.	50-70	100-125	Rotary gear	Hartung type oil relay, governor throttling steam; also automatic nozzle control

Maschinenfabrik Augsburg, Nurnberg.	.....	45-70	Horizontal rotary gear	Hartung type oil relay, governor throttling steam; also automatic steam control on nozzle valves
British Westinghouse Co.	.....	.....	Rotary gear pump	Heavy Hartung type governor direct-connected to throttling valve
Bergmann Electricitäts Werke.	.....	.....	Rotary gear pump	Hartung type governor throttling steam: nozzle valves opened by hand
Belliss & Morcom.	40-50	As high as 90	Double-acting valveless pump, 80-100 r.p.m.	Centrifugal governor, oil relay simply throttling steam supply
Escher, Wyss & Co.	.....	.....	Gear pumps, 1 for relay, 1 for bearing oils	Hartung type, oil relay simple throttling governor
Maschinenfabrik Oerlikon.	Not exceeding 66	Not exceeding 75	Rotary gear pump	Hartung type, oil relay governor, throttling steam supply; main throttle valve also opened by oil relay
Sachsische Maschinenfabrik.	.....	.....	Rotary type	Oil relay governor
Frazer & Chalmers.	.....	.....	.....	.....
Pokorny & Wittekind.	50-80	30-50	.....	Oil relay governor
General Electric Co.	.....	.....	Rotary gear pumps	Vertical centrifugal governor operating oil relay which controls nozzles; nozzle governing
British Thomson Houston Co.	50-60	50-70	Rotary pump	Same as General Electric Co.

TABLE 1—*Continued*

Manufacturer	Oil Data	Type of Glands	Provision for Overspeed Regulation
C. A. Parsons & Co.	.....	Steam-packed labyrinth at shaft	.....
Westinghouse Mach. Co.	Oil pressure on bearings 3 to 4 lb., relay pressure 60 lb.	Water-packed glands with centrifugal impeller	Emergency governor to shut off steam at about 12% overspeed
Willans & Robinson.	Bearing pressure about 10 lb.	Steam-packed labyrinth	Emergency governor for overspeed
Brown Boveri & Co., Richardsons Westgarth & Co.	.....	Steam-packed labyrinth	Emergency governor closes spring controlled throttle valve
Brush Electrical Engrg. Co.	.....	Steam-packed labyrinth; water-packed glands also used	Emergency oil relay control on throttle valve
Erste Brünner M. F. G.	7-10 lb. on bearing	Steam-packed labyrinth	Emergency governor to close spring-loaded throttle valve
Franco Tosi.	10.4 gal. per min. pumped on 300 kw.; 18.4 gal. per min. on 12,000 kw.	Steam-packed labyrinth; water-packed glands also used	Emergency oil relay governor to close main throttle valve at 15% overspeed; unique device
Gebrüder Sulzer.	Bearing oil pressure 20 lb.	Steam-packed labyrinth; labyrinth packing consists of large number thin sheets placed radial, with close clearance on shaft	Oil relay control on steam inlet valve
Melms & Pfenninger. Breitfeld, Danek & Co.	Oil temperature at bearing 105 to 120° F.	Steam-packed labyrinth; exhaust end packing adjustable.	Emergency valve to operate at from 10-15 per cent overspeed
Allis-Chalmers Co.	Bearing pressure 5 lb., relay pressure 30 lb., bearing oil temperature 110° F. best; large quantities pumped	Water-packed glands with centrifugal impeller	Emergency governor to close spring-loaded throttle valve

Allegemeine Electricitäts Gesellschaft.	Oil temperatures at bearings from 175 to 190° F.	Steam-packed labyrinth and carbon rings	Emergency governor to close spring-loaded throttle valve
Maschinenfabrik Augsburg, Nurnberg.	.....	Graphite rings in glands, steam-packed	Emergency governor
British Westinghouse Co.	.....	High-pressure end, labyrinth and water gland; low-pressure end, water gland with centrifugal impeller	Emergency governor closes spring actuated safety valve on steam inlet
Bergmann Electricitäts Werke.	Relay pressure 30 lb.	Steam-packed labyrinth	Emergency governor to close throttle at 10% overspeed
Belliss & Morcom.	Relay pressure 40 lb.; estimate 10 per cent of heat losses to go through oil. design oil pump accordingly	Multi-cellular with carbon rings and steam packed	Emergency governor controls oil relay to shut off steam
Escher, Wyss & Co.	.....	High-pressure end, carbon ring packing; low-pressure, labyrinth	Emergency governor to operate at 8-10% overspeed
Maschinenfabrik Oerlikon.	Oil usually required is 0.02 gal. per kw-min.	High-pressure, graphite; low-pressure, steam-packed labyrinth	Emergency governor closes main throttle valve through oil relay
Sachsische Maschinenfabrik.	.....	High-pressure, carbon; low-pressure, steam-packed labyrinth	Emergency governor to operate at 15% overspeed
Frazer & Chalmers.	.....	High-pressure, metal rings and labyrinth; low-pressure, labyrinth	.....
Pokorny & Wittekind.	From 8-18 gal. per min. circulated according to size	High-pressure, carbon rings and labyrinth; low-pressure, labyrinth	.....
General Electric Co.	Relay pressure about 70 lb., bearing pressure 25 lb.	Carbon ring packing on both ends	Emergency governor to close unbalanced throttle valve
British Thomson Houston Co.	⅓ gal per min. per bearing on 1000 kw. units; 1½ gal. on 5000 kw. unit; same amounts to thrust	Carbon ring packing	Emergency governor to close throttle valve at 15% overspeed

TABLE 1—*Continued*

Manufacturer	Type of Coupling	Additional Notes
C. A. Parsons & Co.	Flexible claw	Simple Parsons turbines, usually very long spindles; large sizes in 2 cylinders
Westinghouse Mach. Co.	Flexible claw	Small sizes simple Parsons; large sizes Curtis-Parsons, double flow in low-pressure, heavy thrust block to take any end thrust
Willans & Robinson.	Flexible claw	.....
Brown Boveri & Co., Richardsons Westgarth & Co.	Flexible claw	Effect on steam consumption of changes in (*a*) vacuum, 5-6% per 1 in. vacuum; (*b*) steam pressure, 1% per 15 lb.; (*c*) superheat, 1% per 11° F.
Brush Electrical Engrg. Co.	Flexible claw	.....
Erste Brünner M. F. G.	Flexible claw	
Franco Tosi.	Flexible claw	Effect on steam consumption of changes in (*a*) vacuum, 7% per 1 in. at 28 in.; (*b*) steam pressure, 1% per 15 lb.; (*c*) superheat, 1% per 12.5° F.; large thrust block in form revolving piston, labyrinth packings; oil admitted both sides, movement shaft opens one side, closes other, making pressures on both different, automatically compensating for end thrust
Gebrüder Sulzer.	Special design of flexible coupling	End thrust all taken up by automatic oil controlled balance piston on end of shaft similar to Franco Tosi's
Melms & Pfenninger, Breitfeld, Danek & Co.	Flexible claw	Effect on steam consumption of changes in (*a*) vacuum 28-29 in., 5%; 27-28 in., 4%; 26-27 in., 3.5%; (*b*) steam pressure 1% per 12 lb.; (*c*) superheat 1% per 10° F.
Allis-Chalmers Co.	Flexible claw	.....

Allegemeine Electricitäts Gesellschaft.	Solid	.....
Maschinenfabrik Augsburg, Nurnberg.	Solid	.....
British Westinghouse Co.	Flexible claw	.....
Bergmann Electricitäts Werke.	Solid	.....
Belliss & Morcom.	Special design of flexible coupling	Flood bearings with great quantity of oil to flush away heat rapidly
Escher, Wyss & Co.	Solid	.....
Maschinenfabrik Oerlikon.	Flexible with leather disks	Blade angles, 20 deg. on stationary blades, 30 deg. on moving blades
Sachsische Maschinenfabrik.	Flexible with leather disks	.....
Frazer & Chalmers.	.....	.....
Pokorny & Wittekind.	Solid	Practically all turbines built for connection to turbo-compressors: effect on steam consumption of changes in (*a*) vacuum live-steam turbines, 5% per in., exhaust steam turbines 10% per in. vacuum; (*b*) steam pressure, 1% per 25 lb.; (*c*) superheat 1% per 12° F.
General Electric Co.	Solid	.....
British Thomson Houston Co.	Solid	Effect on steam consumption of changes in (*a*) vacuum 7% increase per in. above 28 in., 150 lb. steam gage, 8% decrease per in. below 28 in., 150 lb. steam gage; (*b*) steam pressure 1% per 10 lb. around 150 lb. pressure; (*c*) superheat 1% per 10° F.

to compare results, for it is largely dependent on conditions beyond the control of the turbine builder. For instance, if the plant does not contain superheaters, the b.t.u. per kw-hr. will be high. The same will be true of a plant which has a warm cooling-water supply for condensers and consequently carries low vacuum. Yet the turbines may be designed to give a high efficiency ratio under these conditions. In fact, they may be able to utilize the heat available more efficiently than the turbines in another plant with both high superheat and high vacuum.

70 This can be seen in Table 2, in which recent turbine tests have been tabulated. The Brown-Boveri turbine at the Dunstan power plant uses 14,980 b.t.u. per kw-hr. with an efficiency ratio of 68.8 per cent. Yet the Westinghouse City Electric with a lower steam pressure, lower superheat and lower vacuum, has an efficiency ratio of 68.9 per cent, though using 16,925 b.t.u. per kw-hr. The Erste Brünner Vienna turbine requires 16,460 b.t.u. per kw-hr. with 71.8 per cent efficiency ratio. It is therefore apparent that the efficiency ratio alone will express in the best manner the degree to which the designer has approached ideal results in his turbine.

71 The test results in Table 2 were grouped in order to analyze the relative merits of the different types of turbines on the basis of efficiency ratios. The Curtis-Parsons machines built by Erste Brünner hold first place in the list, but are followed closely by others of the same type built by Brown Boveri and Westinghouse Machine Company. The Parsons turbines built by Allis-Chalmers and Brown Boveri also show high efficiencies. The second class in the order of efficiency includes turbines of the Curtis-Rateau and Curtis-Zoelly types, among which the turbines of the A. E. G. and British Westinghouse Company show remarkably good results. The next group includes simple Zoelly and Rateau turbines. The last group comprises straight Curtis types.

72 The superiority of the Curtis-Parsons over the Parsons type is probably due to the reduction in the fluid friction and rotational losses occurring in the first cylinder of the Parsons by the use of a Curtis stage in this section.

73 The Parsons low-pressure sections evidently utilize the heat in the steam only slightly more efficiently than do the impulse turbines. The great surface areas of all disk type turbines which must be whirled in steam, produces losses which are apparently somewhat larger than the combined whirling losses and leakage in the Parsons

drum turbines. Both the Zoelly and Curtis-Rateau types appear to use the steam more effectively in the low-pressure sections than the Curtis alone. Many European engineers hold the opinion that where high economy is to be obtained, the impulse turbine of the Rateau or Zoelly type is superior to the Curtis, though its manufacturing costs are higher. The Curtis-Rateau construction has all the commendable features of impulse turbines and has proved very economical.

74 The results shown in Table 2 are from the best reliable tests that have been made on each type. Objection may be raised that these results do not represent actual operating conditions as under varying loads, nor the average economy of any type of turbine. For instance, the Curtis turbine usually gives a very flat water-rate curve, while the Parsons type is more convex. On the other hand, recent tests on the new Curtis-Parsons types have also shown flat water-rate curves at various loads. It was impossible to compare the various types from this standpoint on account of absence of complete data of such tests.

75 It is interesting to note in Table 2 that the best results have been obtained within the past two years and that these show a considerable increase in efficiency over the earlier turbines.

LOW AND MIXED-PRESSURE TURBINES

76 One of the first low-pressure turbines installed was described by Professor Rateau in his paper on Different Applications of Steam Turbines.[1] The exhaust steam from various non-condensing reciprocating engines around mines was conducted to a regenerator, from which the turbine drew its steam supply.

77 Many low-pressure turbines have been erected since 1904 and have shown very economical results. In some installations it is usual to provide for operation on high-pressure steam when the supply of exhaust steam is insufficient to meet the power demand. Hence, the mixed pressure turbine has been developed. In Europe a Curtis stage is added at the inlet and the live steam passed through this before entering the low-pressure section. The whole of the heat content of the live steam can be effectively utilized by this method. Bleeder turbines are also being built, in which, after partial expansion to some fixed pressure, a portion of the steam is withdrawn from the casing for heating or industrial purposes.

[1] Trans. Am. Soc. M. E., vol. 25, p. 782.

TABLE 2 ECONOMY TESTS OF HIGH PRESSURE STEAM TURBINES

EFFICIENCY RATIOS BASED ON E.H.P. MARKS & DAVIS STEAM TABLES USED

Maker of Turbine	Type	Date of Test	Load-Kw.	R.p.m.	Steam Pressure Lb. Absolute	Temperature at Throttle, deg. fahr.	Vacuum referred to 29.92° Bar	Condenser Pressure, Lb. Absolute	Lbs. of Steam per Kw-Hr.	B.t.u. per Kw-Hr.	Heat Utilized per Lb. of Steam	Heat Available per Lb. of Steam	Efficiency Ratio	Reference
Erste Brünner M. F. G.	Curtis-Parsons	1910	2128	1500	156.2	482	27.89	0.995	13.82	16460	247.0	343.8	71.8	Periodische Mitteilungen
Erste Brünner M. F. G.	Curtis-Parsons		6000	960	184.9	573	28.18	0.854	12.56	15570	271.5	380.7	71.3	Zeit. D.V.D. Ing., 12/10/'10
Erste Brünner M. F. G.	Curtis-Parsons	1910	7442	960	192.0	584	28.18	0.853	12.625	15705	270.2	384.4	70.3	Periodische Mitteilungen
Westinghouse Machine Co.	Curtis-Parsons	1910	9173	1800	181.7	433	27.81	1.032	14.57	16925	234.1	340.2	68.9	Trans. A.S.M.E., vol. 32
Brown Boveri & Cie.	Curtis-Parsons		3053	1360	150.2	505	29.00	0.456	13.01	15990	262.2	385.5	68.0	Dinglers P.J., 6/17/'11
Erste Brünner M. F. G.	Curtis-Parsons	1910	1416	1260	128.2	482	27.60	1.137	15.18	18060	224.6	326.5	68.8	Periodische Mitteilungen
Brown Boveri & Cie.	Curtis-Parsons	1911	1750	1500	176.4	586	27.08	1.392	14.23	17500	239.5	354.8	67.5	Zeit. F.D.G. Turb., 5/30/'11
Brown Boveri & Cie.	Curtis-Parsons	1910	3764	1500	161.2	561	28.77	0.562	13.04	16290	261.5	391.4	66.8	Zeit. F.D.G. Turb., 5/30/'11
Westinghouse Machine Co.	Curtis-Parsons		9830	750	192.2	475	27.22	1.322	15.15	17790	225.2	336.0	67.0	Trans. A.S.M.E., vol. 32
Brown Boveri & Cie.	Curtis-Parsons	1911	1495	3000	200.6	563	26.41	1.720	14.78	17880	230.7	345.5	66.8	Data from Manufacturer
Brown Boveri & Cie.	Curtis-Parsons	1911	1271	3000	172.1	568	27.31	1.278	14.61	17880	233.5	354.3	65.9	Data from Manufacturer
Westinghouse Machine Co.	Curtis-Parsons		11466	750	191.7	484	28.07	0.910	14.45	17210	236.0	360.5	65.5	Trans. A.S.M.E., vol. 32
Erste Brünner M. F. G.	Curtis-Parsons		1250	3000	184.9	573	27.89	0.996	14.32	17680	238.2	373.1	63.9	Zeit. D.V.D. Ing., 12/10/'10
Brown Boveri & Cie.	Curtis-Parsons	1910	3320	1500	180.9	525	29.02	0.440	13.50	16680	252.7	401.3	63.0	Zeit. F.D.G. Turb., 5/30/'11
Brown Boveri & Cie.	Curtis-Parsons		5128	1000	171.2	565	28.52	0.726	14.35	17830	237.7	382.9	62.1	Stodola, 4th ed., p. 449
Breitfield, Danek & Co.	Impulse-Parsons	1909	3585	896	160.7	457	28.32	0.782	16.08	19070	212.0	352.4	60.2	Zeit. D.V.D. Ing., 12/10/'10
Brown, Boveri & Cie.	Parsons	1910	6257	1210	203.7	559	29.02	0.440	11.95	14980	285.5	415.0	68.8	Official Test Report
Allis-Chalmers	Parsons	1908	4300	1800	186.4	484	27.96	0.960	14.02	16690	243.4	355.7	68.4	Sibley Jour. of Eng., 1/11'
Brown Boveri & Cie.	Parsons	1903	3500	1360	156.4	499	28.84	0.532	13.71	16720	248.5	378.6	65.6	Zeit. D.V.D. Ing., 12/10/'10
Brown Boveri & Cie.	Parsons		3000	1360	165.0	625	27.02	1.420	14.75	18433	231.3	359.5	64.3	Die Turbine, 6/20/'11
C. A. Parsons & Co.	Parsons		5164	1200	214.3	509	28.95	0.473	13.18	16140	258.7	402.3	64.3	Stodola, 4th ed., p. 439
Allis-Chalmers	Parsons	1911	3850	1800	164.7	491	27.91	0.983	15.40	18410	221.3	348.3	63.5	Power, 1/2/'12
A. E. G.	Curtis-Rateau	1911	6518	1220	198.7	601	29.28	0.352	11.43	14640	298.4	434.2	68.7	Official Test Report
A. E. G.	Curtis-Rateau	1911	6565	1220	200.2	597	29.18	0.406	11.64	14848	293.0	427.7	68.5	Official Test Report

British Westinghouse	Curtis-Rateau	1911	5066	1500	190.2	552	28.68	0.649	13.00	16100	262.4	391.5	67.0	Electrical Review, 6/23/'11
M. A. N.	Curtis-Zoelly		3584	1500	178.3	569	27.54	1.166	13.99	17190	243.7	361.3	67.5	Data from Manufacturer
Bergmann	Curtis-Rateau	1909	1545	1500	188.5	581	28.59	0.654	12.97	16230	263.0	396.3	66.4	Zeit. D.V.D. Ing., 12/10/'10
Bergmann	Curtis-Rateau	1910	2477	1500	140.0	522	28.81	0.588	13.93	17135	244.8	373.4	65.6	Elec. Zeit., 4/20/'11
A. E. G.	Curtis-Rateau	1908	4239	1500	188.3	662	29.11	0.397	11.97	15620	284.9	439.0	64.9	Stodola, 4th ed., p. 404
British Westinghouse	Curtis-Rateau	1911	2930	1500	210.2	568	28.18	0.894	13.72	16935	248.7	383.3	64.9	Electrical Review, 4/28/'11
A. E. G.	Curtis-Rateau	1907	3169	1500	184.7	592	29.11	0.397	12.74	16230	267.7	425.1	63.0	Trans. A.S.M.E., vol. 32
M. A. N.	Curtis-Zoelly		2507	1500	175.5	460	27.40	1.234	16.24	19020	210.0	334.6	62.8	Data from Manufacturer
Bergmann	Curtis-Rateau	1909	1562	1500	186.8	555	28.33	0.780	14.57	17970	234.1	381.3	61.4	Data from Manufacturer
James Howden & Son	Zoelly	1909	6383	1000	202.7	520	27.33	1.269	14.305	17150	238.5	353.0	67.5	Engineer, London, 10/29/'09
M. A. N.	Zoelly	1910	1400	3000	180.7	554	27.40	1.237	14.21	17310	240.0	356.2	67.4	Zeit. D.V.D. Ing., 12/10/'10
Escher, Wyss & Co.	Zoelly	1910	2052	3000	193.9	585	28.39	0.750	13.04	16290	261.5	392.6	66.6	Zeit. F.D.G. Turb., 2/20/'11
Escher, Wyss & Co.	Zoelly	1910	4189	1000	179.7	557	28.66	0.618	13.30	16520	256.5	391.3	65.5	Zeit. F.D.G. Turb., 2/20/'11
F. Ringhoffer	Zoelly	1908	3000	1000	170.7	470	27.60	1.138	15.52	18278	219.8	339.2	64.8	Zeit. D.V.D. Ing., 12/10/'10
M. A. N.	Zoelly	1910	1250	3000	182.1	582	28.82	0.540	13.09	16500	260.2	404.5	64.4	Zeit. D.V.D. Ing., 12/10/'10
Oerlikon	Rateau	1911	3166	1500	213.9	663	29.25	0.367	11.44	14970	298.2	450.6	66.1	Engineering, 10/20/'10
Escher, Wyss & Co.	Zoelly		5118	1000	133.7	549	27.55	1.161	15.18	18530	224.6	341.6	65.7	Dinglers P.J., 7/15/'11
Escher, Wyss & Co.	Zoelly	1908	5000	1000	166.4	539	26.38	1.736	16.13	19350	211.2	330.4	63.9	Zeit. D.V.D. Ing., 12/10/'10
Escher, Wyss & Co.	Zoelly		3540	1500	155.1	469	28.21	0.838	15.07	17940	226.3	349.5	64.8	Dinglers P.J., 7/15/'11
Escher, Wyss & Co.	Zoelly	1910	1641	3000	221.0	672	27.91	0.985	13.08	16775	260.6	406.5	64.1	Zeit. F.D.G. Turb., 2/20/'11
Escher, Wyss & Co.	Zoelly	1910	1235	3000	176.8	451	28.39	0.750	15.35	18156	222.3	357.8	62.2	Zeit. F.D.G. Turb., 2/20/'11
British Thomson-Houston	Curtis	1911	2987	1500	154.7	505	26.75	1.557	15.96	18960	213.7	321.2	66.5	Engineering, 10/20/'11
Gen. Elec. Co.	Curtis		3464		210.0	513	28.75	0.575	13.62	16620	250.4	393.4	63.6	Trans. A.S.M.E., vol. 32
British Thomson-Houston	Curtis	1909	2500	1500	126.5	414	28.47	0.711	15.92	18590	214.0	336.1	63.7	Zeit. D.V.D. Ing., 12/10/'10
A. E. G.	Curtis	1906	3000	1500	191.3	590	29.05	0.427	12.79	16240	266.6	420.4	63.4	Zeit. D.V.D. Ing., 12/10/'10
A. E. G.	Curtis	1909	2236	1500	191.6	654	29.34	0.284	11.77	15450	289.8	455.8	63.6	Zeit. D.V.D. Ing., 12/10/'10
Gen. Elec. Co.	Curtis		8880		192.5	487	28.02	0.933	15.05	17965	226.7	359.5	63.1	Trans. A.S.M.E., vol. 32
British Thomson-Houston	Curtis	1911	1541	1500	149.7	365	27.97	0.956	17.46	19720	195.3	320.2	61.0	Engineering, 10/20/'11
Gen. Elec. Co.	Curtis		10816	750	190.0	525	29.39	0.260	12.90	16135	264.5	427.3	61.9	Trans. A.S.M.E., vol. 32
Gen. Elec. Co.	Curtis		5095		185.1	554	29.40	0.255	12.71	16090	268.4	436.0	61.6	Trans. A.S.M.E., vol. 32
British Thomson-Houston	Curtis	1911	1221	3000	134.7	448	27.16	1.353	17.75	20690	192.2	314.0	61.2	Engineering, 10/20/'11
Gen. Elec. Co.	Curtis	1910	8775	750	194.0	451	27.95	0.956	15.95	18720	213.8	350.8	61.0	Trans. A.S.M.E., vol. 32

References: Zeit. D.V.D. Ing.—Zeitschrift des Vereines Deutscher Ingenieure; Zeit. F.D.G. Turb.—Zeitschrift für das Gesamte Turbinenwesen; Dinglers P.J.—Dinglers Polytechnisches Journal; Elec. Zeit.—Electrotechnische Zeitschrift.

78 Low-pressure turbines are frequently installed to use the exhaust steam of reciprocating engines without regenerators. In this case the generators are sometimes tied together electrically and the turbines are only fitted with an overspeed governor. Messrs. Stott and Pigott showed the results that could be obtained from such a combination in their paper, Test of a 15,000-Kw. Steam-Engine-Turbine Unit.[1] Such turbines are usually installed in stationary work, only when the reciprocating engines are already in service. The high-pressure turbine in a new plant requires less floor space, has less complicated machinery, is cheaper in first cost and in maintenance, and approaches, if it does not equal, the economy to be derived from the combination unit in every-day service. Nevertheless, there have been a number of combined engine and turbine plants recently installed in England which have proved very satisfactory.

79 It is probable that low-pressure turbines will be installed in the near future in large gas engine stations to utilize the waste heat in the gas engine exhausts.

TURBO-COMPRESSORS

80 Turbo-compressors have some decided advantages over reciprocating compressors, such as smaller floor space, absence of inlet and discharge valves, low cost of upkeep and no internal lubrication. They are being manufactured quite extensively in Europe and have been introduced in America by the General Electric Company, one of whose turbines was described by Mr. R. H. Rice in his paper, Commercial Application of the Turbine Turbo-Compressor,[2] read at the Pittsburgh meeting of the Society.

81 Turbo-compressors are built either with curved or radial impeller blades which discharge the air into smooth expanding diffusor channels to convert the velocity energy of the current into pressure. Usually guides are provided to direct the air into the entrance of the next stage without eddies. All passages are made as smooth as possible with no abrupt bends or turns, and all walls are water-cooled. The air is prevented from leaking back from stage to stage by labyrinth packings such as are used in impulse turbines.

82 European builders have installed turbines to deliver air at as high pressures as 130 lb. per sq. in. gage and are prepared to furnish them up to 180 lb. discharge pressure.

[1] Trans. Am. Soc. M. E., vol. 32, p. 69.

[2] The Journal Am. Soc. M. E., March 1911, p. 301.

83 Turbo-blowers have also been built to deliver large volumes of air at low pressure, such as are required in furnace work. These units have no water-cooled jackets. The first difficult problem encountered in the construction of turbo-blowers or compressors was the provision of a suitable governing device for the unit. However, several ingenious and satisfactory arrangements have recently been developed and it is probable that this difficulty will soon be overcome completely.

84 The efficiency of turbo-compressors with water-cooling is defined as the ratio of the power required to compress the given quantity of gas isothermally to the power consumed at the compressor coupling in the actual compression. This efficiency in well designed units with discharge pressures between 60 and 150 lb. should fall within the limits of 60 and 70 per cent. The best results noted up to the present time were obtained on a turbo-compressor built by Pokorny and Wittekind for the Victoria Falls Power Company in South Africa, which on official test showed an efficiency of 67.7 per cent.

85 When there is no water-cooling provided, the efficiency of a turbo-blower is expressed as the ratio of the power required to compress the given quantity of air adiabatically to the power actually expended at the compressor coupling. This efficiency, depending on the size of the blower, should fall between 70 and 80 per cent as a maximum. An efficiency of 78 per cent has been obtained on official tests of a Rateau turbo-blower built by Kuhnle, Kopp and Kausch and is probably the best result obtained up to the present time on this type of compressor.

86 It can thus be seen that in so far as efficiency is concerned, the turbo-compressor is equal to the average reciprocating compressor. It seems probable that turbo-compressors and blowers will be used to an increasing extent, largely on account of their low first cost and operating costs as compared with steam reciprocating units.

87 The high thermal efficiency of the gas-driven blowing engine exceeds that possible in a turbo-compressor unit, so that the former will continue to be used in blast-furnace work.

TURBO-DRIVEN PUMPS

88 The steam turbine is an ideal source of power to drive centrifugal pumps, especially when it is necessary to lift against high heads. Hence it has been installed in several places for city fire service, using lake or river water in high-pressure mains. The

efficiency of such centrifugal pumps usually ranges from 65 to 80 per cent, so that, in spite of the high efficiency of the turbine itself, the combined set will not give as good economy as a high grade reciprocating pumping engine. However, its first cost is low, it requires no internal lubrication, takes up very little floor space, and has no valves to require examination or renewals.

GEARED TURBINES

89 Attempts have been made to adopt steam turbines for direct connection to continuous current generators and other slow-speed machinery. A steam turbine to be economical must be a high-speed machine, and hence its use with slow-speed machinery has not proved entirely satisfactory. Dr. De Laval adopted spur gearing as a means of reducing speeds on his first simple impulse turbines, and builders of this type still use this construction. It is only within the last few years that attempts have been made to apply gearing as a means of speed reduction on other types of turbines. The Westinghouse Machine Company are now manufacturing direct-current turbo-generators with the Melville-MacAlpine reduction gearing between turbine and generator. This gear is also being built for use in marine work to drive slow-speed propellers. The gear wheels in this construction are carried on a floating frame so that the teeth may be always in correct alignment. One set of such gearing showed under test an efficiency of 98.5 per cent.

90 C. A. Parsons & Co. have built several notable reduction gears for steam turbines, in which the floating frame idea was omitted. A mixed pressure steam turbine of 750 b.h.p. is now in use driving a three high set of rolls through gearing at the Calderbank Steel Works near Glasgow, Scotland. A flywheel is placed on the same shaft as the driven gear and thus takes the shock off the turbine when a billet enters the rolls. In 1909 the Parsons Marine Steam Turbine Company installed a geared turbine of 1000 h.p. in the S. S. Vespasian of 4000 tons. Extensive experiments were carried out and it was found that the efficiency of this gearing, which had no floating frame, ranged between 98 and 99 per cent. After a year's operation, in which the ship covered 20,000 miles, tests were again made on the gearing with equally good results. On examination no appreciable signs of wear could be noted on the gear teeth, which were made of mild chrome nickel steel and were flooded with oil.

91 It is perfectly feasible to adopt the steam turbine through gearing to belt and rope drives when these are required in large powers. Under certain conditions, where fuel costs are high and water for condensation is plentiful, such an installation would prove an economical investment in place of reciprocating engines. The direct turbine drive would then come into competition with the motor drive. The losses in lineshafting can be greatly reduced by the use of ball or roller bearings so that the turbine drive may prove very economical in some instances as compared with individual motor drives.

92 The use of geared turbines in large sizes has been in the nature of an experiment until quite recently. Judging from the results obtained in recent installations, their commercial success seems now assured. In the design of these units the tendency will be to simplify details. The success of the Parsons gears on the S. S. Vespasian should encourage designers to do away with any special devices to secure alignment and to provide simply accurately cut gears properly meshed and running in a flood of oil.

MARINE TURBINES

93 All the standard types of turbines have now been adopted for marine service in driving screw propellers, either direct-connected or through gearing. The design of direct-connected turbines is complicated by the fact that the speed of screw propellers must necessarily be low as compared with the most favorable speeds for economical steam-turbine operation. Hence these turbines require large spindle diameters and massive construction and yield correspondingly poor steam economies, especially at slow speeds. These units are usually built with the power divided between two or more shafts connected to high and low-pressure cylinders. With geared equipments the turbines can be operated at their most efficient speeds, while the gears can be so designed that the propeller also runs at its most economical speed.

94 Reversing is made possible by suitable blading in the low-pressure ends of the main units into which live steam is admitted when desired. When the turbine is running forward, this blading revolves in vacuum and consumes but little power.

95 Combined types of turbines are also being introduced in marine installations. A recent Curtis design includes a drum impulse section. M. Zoelly now uses Curtis stages in his high-pressure section, but with steam velocities not exceeding 1300 ft. per sec.,

obtained by converging nozzles only, and drum impulse construction on the low-pressure ends. Several other European builders have also used Curtis stages in the high-pressure portion.

96 Reciprocating engines exhausting into low-pressure turbines have been installed in several ships, the most notable of which is the Olympic, and have shown very satisfactory results. In this case the engines are used for reversing.

97 Many schemes have been proposed to install turbo-generators of central station type on shipboard and to operate the propeller shafts by means of large slow-speed induction motors. Marine engineers object to this arrangement on account of the dangers accompanying the use of such electrical machinery and auxiliaries in marine service. This objection seems to be due largely to inexperience with electrical machinery, as the essential conditions of operation in marine work do not differ greatly from those under which many electrical machines operate satisfactorily in land practice.

98 Turbo-driven lighting sets and other auxiliaries are being used in increasing numbers on shipboard, owing largely to the high efficiencies which may be obtained, to the small floor space required, and to the light weight of the units.

99 Recent orders for turbine-driven steamships abroad include some interesting equipments. The Canadian Pacific Railway has ordered two boats with four screws and with Parsons turbines. The two outside screws will have high and intermediate pressure turbines respectively, while the two center screws will be connected to low-pressure turbines. Two twin-screw passenger boats of 5000 h.p. with Parsons turbines have been ordered for the Southampton-Havre service. Each set consists of a high-pressure and a low-pressure turbine geared individually to its own propeller shaft. The British Government has ordered two twin-screw destroyers also to use Parsons geared turbines, totalling 14,000 h.p. per ship, or 7000 h.p. per gear. The United States Government has placed an order with the Westinghouse Machine Company for a gear equipment on one of its colliers.

100 Geared turbine equipments are thus making rapid headway on account of the high efficiency of the combination and the resultant favorable steam consumption obtained.

101 At the present time about 90 per cent of the marine turbines built have been of the Parsons type. Here again the inefficiency of the Parsons high-pressure sections has become apparent, so that it is probable that a construction similar to Zoelly's high-pressure end

will be introduced in this section. Sir Charles A. Parsons is quoted as saying: "In the low-pressure blades of the Mauretania the leakage was practically nothing and their efficiency was about 85 per cent." [1] Under such conditions it does not seem probable that much higher efficiency can be obtained by use of other constructions than the Parsons reaction type in the low-pressure section. It is reported that the low-pressure blading of Parsons turbines in ships of the United States Navy has given considerable trouble, and also that the turbines need more careful handling when starting up than do impulse turbines.

TREND OF TURBINE DEVELOPMENT

102 The cost of manufacture is a very important item in determining the future development of the steam turbine. Types such as the original Parsons and the Rateau, while inherently of very high efficiency, have too high manufacturing costs to compete with the newer combined types.

103 The writer offers as his opinion that the combined types, such as the Curtis-Parsons, the Curtis-Rateau and also the Curtis-Rateau-Parsons, previously described, will very soon supersede the simple types. It is probable that the Curtis turbine will eventually be built only in horizontal units and will gradually be modified to a Rateau or even a drum impulse construction in the low-pressure sections. The freedom from close adjustment in impulse turbines and the recent improvements in blading materials will greatly increase the use of this type, although Curtis-Parsons turbines are said to be cheaper to manufacture. In actual operation, it is an open question among engineers whether the reaction turbine has a higher commercial efficiency than the impulse type, and hence buyers usually consider first cost and personal preference only.

104 Turbines will probably be made shorter with very stiff shafts. With this construction many of the earlier blading troubles will disappear. But the peripheral speeds will also be increased and this will involve the development of suitable blading material and methods of holding blades that will satisfy these new requirements. Recent results seem to indicate that improved efficiency may be looked for with increased blade speeds.

105 Several impulse turbines have been built recently in Europe where the expansion was not complete in the nozzle, so that a portion of the expansion took place in the first moving blades. Some

[1] Engineering, Oct. 27, 1911.

large Curtis turbines recently installed in America are said to have Parsons blading in the last stage. These developments would indicate a movement to introduce reaction principles in impulse turbines, and further illustrate the tendency to merge types.

106 The hope of further improvement in efficiency lies in extensive study, particularly of the action of the steam during its passage through the moving and stationary blades, of the effect of form of blades, passages and casings and of various forms of baffles and balance pistons to prevent leakage. Such research work has not been carried out up to the present time by most manufacturers, largely on account of the extreme care and heavy expense involved in such tests. The present state of development has been largely one of cut and try. The increasing competition of the gas engine and the possible development of a satisfactory gas turbine will force manufacturers to develop their turbines to the greatest degree of economy.

107 With regard to detail, simplicity will be the leading consideration. With the introduction of Curtis high-pressure stages, nozzle governing will undoubtedly be used to an increasing extent, though the results obtained by Westinghouse, by Zoelly, and by Bergmann with simple throttling governors, raise a question as to whether the additional complication of nozzle governing will pay. Oil relays will probably replace all other systems of governing on account of their simplicity and reliability. The simple and efficient centrifugal oil pump governor of Sulzer appears to be an improvement of considerable moment, and will probably receive extensive use.

108 With the development of suitable gearing for steam turbines, their field of application has been greatly increased and turbines will shortly be used for purposes which engineers today would consider them utterly unfit. Low-pressure turbines will continue to be installed in plants where reciprocating engines are still in operation and also where large quantities of waste heat are available. The low and mixed pressure types of turbines will find a very extended use in connection with heating systems, evaporators, etc.

109 The development of the past ten years has been truly marvellous. No great gain in thermal efficiency seems possible, so that future improvements will be largely along the line of detail construction and modification.

110 The addresses of the firms referred to in Table 1 are as follows: C. A.

Parsons & Co., Newcastle, England; Westinghouse Machine Co., East Pittsburgh, Pa.; Willans & Robinson, Rugby, England; Brown Boveri & Co., Baden, Switzerland; Richardsons Westgarth & Co., Hartlepool, England; Brush Electrical Engineering Co., Loughborough, England; Erste Brünner, Brünn, Austria; Franco Tosi, Legnano, Italy; Gebrüder Sulzer, Winterthur, Switzerland; Melms & Pfenninger, Munich, Germany; Breitfeld Danek & Co., Prague, Austria; Allis-Chalmers Co., Milwaukee, Wis.; Allegemeine Electricitäts Gesellschaft, Berlin, Germany; Maschinenfabrik Augsburg Nürnburg, Nürnburg, Germany; British Westinghouse Co., Manchester, England; Bergmann Electricitäts Werke, Berlin, Germany; Belliss & Morcom, Birmingham, England; Escher Wyss & Co., Zurich, Switzerland; Maschinenfabrik Oerlikon, Oerlikon, Switzerland; Sachsische Maschinenfabrik, Chemnitz, Germany; Frazer & Chalmers, Erith, England; Pokorny & Wittekind, Frankfort-Bockenheim, Germany; General Electric Co., Schenectady, N. Y.; British Thomson Houston Co., Rugby, England.

The writer wishes to express his thanks to manufacturers of steam turbines in America and in Europe, who liberally provided him with the information on which this paper is based, both during personal visits to their works and by correspondence.

PROBLEMS IN NATURAL GAS ENGINEERING

By Thos. R. Weymouth

ABSTRACT OF PAPER

In the production and transportation of natural gas many problems are encountered requiring special applications of engineering principles. It is the purpose of this paper to point out the most important of these and to outline the methods of solution. A brief discussion is given of the properties of natural gas, including a table of analyses of gases produced in the principal gas fields of the United States, together with a formula connecting the heating value of the natural gas with its specific gravity. This is followed by the development of the original formulae for the flow of gas in pipe lines, the power required for compression and the storage capacity of lines, and examples are worked out showing the general method of design of a transmission system.

PROBLEMS IN NATURAL GAS ENGINEERING

By Thos. R. Weymouth, Oil City, Pa.

Member of the Society

The extensive increase in the production and distribution of natural gas in recent years has given rise to engineering problems of more or less unique interest. For a considerable period after natural gas had become an important factor in industrial life it was handled almost entirely by rule of thumb methods, but with the advent of large compressing stations and the enforced realization that the gas supply is not by any means unlimited, the application of engineering principles became essential, not only to secure greater economy, but also to cope successfully with the growing complexity of the problems encountered. The object of this paper is to present the most important of these problems and to outline the methods used in solving them. First, however, it is believed that a brief account of the characteristics of natural gas as it is found in America, will not be out of place.

PROPERTIES OF NATURAL GAS

2 *Origin and Composition.* Natural gas and petroleum are closely related and it is generally believed that they were generated in the earth by one and the same process. What this process is, or may have been, is a question of pure speculation among scientists, two principal theories having been advanced to explain it, both based upon geological observations.

3 The first hypothesis, the so-called chemical, or inorganic theory, holds that the gas is constantly being formed by the action of water on the carbides of different metals in the far inner regions of the earth, thus producing the hydrocarbons forming the principal constituents of natural gas. These hydrocarbons are then forced outwards toward the earth's surface by the high pressures generated until they finally lodge in the porous rocks in which we now find them.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

4 The second, or organic theory, maintains that the hydrocarbons were formed by the partial decomposition under water, of animal or vegetable matter in the rock strata where the gas is now found.

5 There are many arguments in support of each theory, as it is known that hydrocarbons can be produced by either of the methods suggested. This being so, it seems reasonable to suppose that both agencies were at work and that each may claim its share of the credit for the production of the vast quantities of gas obtained throughout the world.

6 By whatever manner produced, however, the fact remains that hydrocarbons form the principal constituents of natural gas as found in America, although other gases are present in widely varying, but relatively small proportions.

7 The analyses of gases found in pools covering practically the whole productive area of the United States are given in Table 1, which was compiled from various sources, the principal of which were the reports of the Geological Surveys of West Virginia, Vol. 1-a, 1904, and of Kansas, Vol. 9. These reports give very complete discussions of the theories mentioned, and describe in detail what the author believes to be the most reliable analytical work done on natural gas, namely, that of Prof. F. C. Phillips on Pennsylvania Natural Gas, in 1887, and that of Cady and McFarland on the Gases of Kansas, in 1906.

8 The table shows what an apparent lack of uniformity exists in the constitution of natural gas from different fields; but the most noticeable feature is the great preponderance of hydrocarbons, especially of the methane, or paraffin series. This group has the general chemical formula C_nH_{2n+2}, its chief member being methane, or marsh gas (CH_4). Methane occurs in all natural gas in proportions varying from about 14 per cent up to 97 per cent. It possesses high heating value but has practically no illuminating properties.

9 The second member of the series, ethane (C_2H_6), occurs in percentages ranging from 0 up to 40 or more. This gas is of much higher fuel value than methane, is considerably heavier, and possesses greater illuminating properties. Thus the heating value, specific gravity, and illuminating value of the gas in which it occurs are all raised by the presence of ethane.

10 The higher members of this series, propane (C_3H_8) and butane (C_4H_{10}), are present in dry gas only in minute quantities, if at all, and they are consequently of small importance.

11 Of the olefine series, or so-called "illuminants," having the general formula C_nH_{2n}, the only member present in measureable quantity is the leading one of the group, ethylene (C_2H_4), and this only in a fraction of 1 per cent.

12 Carbon monoxide (CO) is occasionally reported, although in such small proportion as to add very little to the fuel value of the gas.

13 Hydrogen has been reported in many analyses of natural gas, but, as has been pointed out by Phillips, Cady and McFarland, these indications were probably due to erroneous deductions from experimental results, and it is extremely doubtful if free hydrogen is ever present.

14 Hydrogen sulphide (H_2S) is found in the natural gas from a few isolated areas, but is not at all general. Its presence is indicated by a distinctive odor. At most, it exists in but a small fraction of 1 per cent and is, therefore, relatively unimportant.

15 The above list comprises all the gases ever found in natural gas that contribute to its heating value, and it will be seen that the only ones of any practical importance are methane and ethane, the first two members of the paraffin series.

16 In addition to the above, there are always found a number of inert gases, chief of which is nitrogen (N), which ranges in proportion from less than 1 per cent to almost 83 per cent in the various gases shown in Table 1. The majority of natural gases, however, that exist in any considerable quantity, or come from any great depth, seldom exceed 10 per cent in nitrogen content, and usually range from 4 to 7 per cent. Nitrogen is a non-reactive, elementary gas that acts merely as a diluent of the natural gas, reducing its fuel value.

17 Next in importance is carbonic acid gas (CO_2), which is found in proportions ranging from 0 up to about 4 per cent. It acts in the same manner as nitrogen, by reducing the effective heating value of the gas.

18 Oxygen has frequently been reported in varying quantities, but the probability is that it is present in traces only, and that the apparent high percentages sometimes reported were due to the samples having become contaminated by air.

19 The work of Cady and McFarland has shown that in all probability the element helium is a constituent of nearly all natural gases, together with other members of the argon group. Helium was

TABLE 1 ANALYSES OF GASES FROM THE WHOLE PRODUCTIVE AREA OF THE UNITED STATES

Location (Kansas Gases)	Year	O_2	CO_2	C_2H_4	CO	CH_4	C_3H_8	CH_4 C_2H_6	H_2	H_2S	He	N_2	B.T.U. by Analysis	B.T.U. by Equation [4]	Specific Gravity
*Dexter	1905	0.20	0.00	.....	0.00	14.85	0.41	15.22	trace	.....	1.84	82.70	139.7	82.6	0.8935
*Dexter (Greenwell Well)	1906	0.10	0.00	0.00	0.00	14.33	1.06	15.39	trace	.....	1.64	82.87	245.5	89.6	0.8977
*Eureka, new field	1906	0.10	0.20	0.00	0.00	51.80	0.00	51.80	0.00	.....	1.50	46.40	465.0	427.6	0.7430
*Eureka, town supply	1906	0.50	0.20	0.00	0.00	51.40	0.00	51.40	0.00	.....	1.50	46.40	461.0	424.6	0.7451
*Fredonia	....	trace	0.61	0.12	0.00	82.25	0.00	82.25	0.00	.....	0.616	16.40	739.8	729.6	0.6253
*Elmdale	1906	0.30	0.15	0.55	0.00	78.60	7.71	86.31	0.00	.....	0.56	12.13	836.1	826.1	0.6441
*Moline	1906	0.00	0.54	.....	0.00	74.10	0.00	74.10	trace	.....	0.51	24.85	665.0	652.2	0.6599
Burlington	1906	0.00	0.00	0.20	0.00	85.50	3.20	88.70	0.00	.....	0.495	10.60	820.9	810.3	0.6113
New Albany	1906	0.00	0.20	0.25	0.00	89.10	0.00	89.10	0.12	.....	0.49	9.84	804.0	794.7	0.5944
*Lawrence, deep well	1906	trace	0.92	0.00	0.00	81.40	0.00	81.40	0.00	.....	0.46	17.22	730.0	723.9	0.6318
*Olathe	1906	trace	0.00	0.10	0.00	84.40	0.00	84.40	0.00	.....	0.40	15.10	758.5	746.2	0.6148
Garnett	1906	trace	0.20	0.16	0.00	94.30	0.36	94.66	0.00	.....	0.37	4.61	855.1	848.9	0.5748
*Eudora	....	0.31	0.62	0.00	0.00	88.60	0.00	88.60	0.00	.....	0.27	10.20	795.0	792.3	0.6021
Parsons (Wilson County Gas)	....	trace	0.72	0.00	0.00	91.90	3.37	95.27	0.00	.....	0.27	3.74	878.8	877.7	0.5902
Buffalo	1906	trace	0.00	0.00	0.11	96.20	0.78	96.98	0.18	.....	0.27	2.46	876.3	873.5	0.5656
Altoona	1906	0.00	0.92	0.00	0.00	92.00	2.85	94.85	0.00	.....	0.263	3.97	870.5	873.1	0.5911
*Augusta	1906	0.00	0.00	0.77	0.00	79.10	7.44	86.54	0.00	.....	0.25	12.44	840.0	831.4	0.6430
Chanute	1906	0.10	0.00	0.00	0.00	94.70	0.00	94.70	0.00	.....	0.24	4.96	850.0	844.7	0.5731
Moran	1906	0.20	0.30	0.35	0.20	92.00	0.00	92.00	0.39	.....	0.214	6.35	832.9	831.8	0.5829
Peru	1906	0.10	0.51	0.00	0.00	81.70	7.60	89.30	0.00	.....	0.19	9.39	854.2	852.5	0.6309
Iola	1906	0.23	0.00	0.00	0.00	94.50	0.00	94.50	trace	.....	0.183	5.08	848.0	843.9	0.5746
Lawrence pipe line, April 16	1906	trace	0.20	.....	0.00	98.06	0.00	98.06	trace	.....	0.17	1.57	880.0	878.3	0.5605
Pipe line, October 23	1905	0.12	0.00	.....	0.00	98.00	0.00	98.00	0.00	.....	.....	1.88	879.0	879.0	0.5615
Pipe line, December 12	1906	0.24	1.94	0.00	0.00	94.30	0.75	95.05	0.00	.....	0.17	2.60	858.0	869.8	0.5868
Arkansas City	....	0.20	0.10	0.10	0.00	81.10	11.95	93.05	0.00	.....	0.159	6.39	920.0	915.8	0.6302
Blackwell, Okla	1906	0.00	0.00	0.61	0.00	83.40	10.31	93.71	0.33	.....	0.16	5.19	922.4	918.4	0.6247
Humboldt	1906	0.10	0.00	0.81	0.00	94.00	1.97	95.97	0.00	.....	0.14	5.19	887.3	880.8	0.5996
Iola	1906	0.40	0.70	0.00	0.30	91.50	0.00	91.50	0.00	.....	0.132	6.97	822.0	826.8	0.5917
Erie	1906	0.22	0.33	0.30	0.00	90.30	4.26	94.56	0.00	.....	0.13	4.45	882.3	881.8	0.5971
Bonner Springs	....	0.10	0.00	0.00	0.00	97.18	0.00	97.18	0.25	.....	0.104	2.36	871.7	870.4	0.5616
Altamont pipe line, Liberty and Coffeyville	....	0.00	0.92	0.61	0.00	95.70	0.00	95.70	0.00	.....	0.08	2.69	868.0	872.1	0.5751
Caney	....	0.15	0.81	0.10	0.00	92.40	0.00	92.40	0.00	.....	0.08	6.46	830.5	833.1	0.5886
Sheffield, Mo	....	0.20	0.83	0.50	0.10	92.90	0.00	92.90	0.00	.....	0.041	5.43	840.7	845.9	0.5869
Kansas City, Mo., 2416 Tracey Avenue	....	0.10	0.60	1.20	0.20	87.20	7.03	94.23	0.00	.....	0.013	3.65	912.2	917.8	0.6141
Paola	1906	0.40	0.70	0.00	0.00	98.00	0.00	98.00	0.00	.....	0.0093	0.88	879.0	884.7	0.5655
Topeka pipe line, December 22	1907	0.20	1.53	0.00	0.00	94.56	0.00	94.56	0.00	.....	.....	3.71	849.0	858.1	0.5841
Lawrence pipe line, December 23	1907	trace	1.85	0.00	0.00	94.47	0.00	94.47	0.00	.....	.....	3.68	848.0	859.6	0.5861
Gas from coal mine, Scranton, Pa	....	0.92	1.06	0.39	0.13	94.2	0.00	94.20	0.00	.....	trace	3.31	851.1	861.0	0.5842
Lawrence pipe line, July 11	1908	0.25	0.81	0.38	0.81	93.50	0.00	93.50	0.00	.....	.....	4.25	847.2	877.6	0.5848
*W. Bloomfield, N. Y	1870	0.23	10.11	2.94	....	82.41	.....	.....	....	.....	4.31	.....	782.3	855.6	0.6819

*Olean, N. Y.	1885	2.00	.....	1.00	0.50	96.50	.....	.....	.....	.....	.....	.....	881.3	890.8	0.5703
Fredonia, N. Y.	1887	....	0.41	.....	....	46.43	43.62	90.05	.....	.....	.....	9.54	1102.5	1110.2	0.8078
Baldwinsville, N. Y., (Trenton)	1899	trace	.....	0.25	0.95	98.40	.....	98.40	trace	.....	.....	0.40	889.3	900.4	0.5596
Morgantown, W. Va. (Big Injun)	1904	0.20	0.006	0.40	0.40	80.94	14.60	95.54	trace	.....	.....	3.46	965.4	969.1	0.6427
Fairmount, W. Va. (Bayard)	1904	0.20	0.100	0.20	0.40	81.60	14.09	95.69	0.20	.....	.....	3.21	961.2	965.6	0.6379
Shinnston, W. Va. (Big Injun)	1904	0.20	0.000	0.40	0.40	79.95	15.09	95.04	0.00	.....	.....	3.96	964.7	968.5	0.6469
Shinnston, W. Va. (Gordon)	1904	0.10	0.00	0.20	0.40	80.85	14.88	95.73	0.10	.....	.....	3.47	966.7	969.6	0.6420
Lumberport, W. Va. (5th. sand)	1904	0.30	0.10	0.10	0.40	80.70	14.35	95.05	0.10	.....	.....	3.95	955.8	958.6	0.6430
Shinnston W. Va. (50 ft.)	1904	0.30	0.00	0.20	0.50	86.48	7.65	94.13	0.00	.....	.....	4.87	902.0	905.9	0.6148
Sheffield, Pa.	1887	trace	0.30	.....	....	70.55	20.09	90.64	.....	.....	.....	9.06	953.1	951.4	0.6908
Kane, Pa.	1887	trace	0.20	.....	....	68.96	21.05	90.01	.....	.....	.....	9.79	954.0	950.6	0.6975
Wilcox, Pa.	1887	trace	0.21	.....	....	72.64	17.74	90.38	.....	.....	.....	9.41	934.5	931.2	0.6803
Speechley, Pa.	1887	trace	0.05	.....	....	67.89	27.55	95.44	.....	.....	.....	4.51	1048.0	1046.3	0.7057
Lyons Run, Murraysville, Pa.	1887	trace	0.28	.....	....	97.60	0.10	97.70	.....	.....	.....	2.02	877.1	877.2	0.5647
Raccoon Creek, Pa.	1887	trace	trace	.....	....	73.72	16.37	90.09	.....	.....	.....	9.91	922.0	918.9	0.6745
Baden, Pa.	1887	trace	0.41	.....	....	70.66	16.61	87.27	.....	.....	.....	12.32	885.5	882.4	0.6879
Houston, Pa.	1887	trace	0.44	.....	....	65.71	18.55	84.26	.....	.....	.....	15.30	899.0	895.5	0.7109
Murraysville, Pa.	1894	trace	0.20	.....	....	93.33	2.07	95.40	.....	.....	.....	4.40	870.0	868.7	0.5833
Pittsburg, Pa., (Exhibition Grounds)	1894	trace	0.52	.....	....	87.56	4.62	92.18	.....	.....	.....	7.30	859.1	860.1	0.6108
Creighton, Pa.	1894	....	3.64	.....	....	86.87	9.49	96.36	.....	.....	.....	0.00	930.8	957.5	0.6340
Pittsburg, Pa. (Painter Well)	1894	....	0.40	.....	....	92.67	6.23	98.90	.....	.....	.....	0.70	930.9	933.7	0.5900
Allegheny City, Pa. (Salt Well)	1894	....	0.30	.....	....	87.53	5.07	92.60	.....	.....	.....	7.10	866.4	865.5	0.6101
Gambier, O.	1875	0.80	0.30	.....	0.50	81.40	12.20	93.60	.....	0.650	.....	4.80	924.5	932.7	0.6414
Findlay, O.	.....	0.34	0.26	0.30	0.50	92.61	.....	92.61	2.18	0.20	.....	3.61	844.1	851.7	0.5665
Thurston, O. (Clinton)	.....	0.15	0.25	0.30	0.15	90.48	.....	90.48	0.55	.....	.....	8.12	817.9	818.3	0.5894
Fostoria, O.	.....	0.35	0.20	0.20	0.55	92.84	.....	92.84	1.89	0.15	.....	3.82	843.7	851.1	0.5678
Findlay, O.	.....	0.39	0.25	0.35	0.41	93.35	.....	93.35	1.64	0.20	.....	3.41	849.6	856.2	0.5682
St. Marys, O.	.....	0.35	0.23	0.20	0.44	93.85	.....	93.85	1.74	0.21	.....	2.98	852.0	859.7	0.5652
Cleveland, O.	1894	....	0.20	.....	....	76.77	16.73	93.50	.....	.....	.....	6.30	955.6	953.5	0.6620
Butler, O.	....	0.05	0.00	0.40	0.00	70.00	16.75	86.75	0.27	.....	0.15	12.38	901.6	895.0	0.6857
Muncie, Ind. (Trenton)	....	0.35	0.25	0.25	0.45	92.67	.....	92.67	2.35	0.15	.....	3.53	844.5	850.9	0.5647
Anderson, Ind. (Trenton)	....	0.42	0.26	0.49	0.73	93.07	.....	93.07	1.86	0.15	.....	3.02	850.5	860.7	0.5676
Kokomo, Ind. (Trenton)	....	0.30	0.29	0.30	0.55	94.16	.....	94.16	1.42	0.18	.....	2.80	856.2	864.4	0.5668
Marion, Ind. (Trenton)	....	0.55	0.30	0.15	0.60	93.58	.....	93.58	1.20	0.20	.....	3.42	848.6	858.8	0.5720
Marion, Ind. (Trenton)	....	0.00	0.73	0.86	0.00	77.40	14.18	91.58	0.00	.....	0.167	6.66	933.2	933.1	0.6592
Kokomo, Ind. (Trenton)	....	....	0.40	.....	....	88.91	4.69	93.60	....	.....	.....	6.00	871.3	872.0	0.6046
Oregon	1904	....	2.40	.....	....	88.10	....	88.10	....	.....	.....	9.50	790.5	804.3	0.6159
*Fort Worth, Tex.	1910	0.20	0.00	0.30	0.30	55.90	5.50	61.40	1.00	.....	.....	36.80	596.8	585.6	0.7320
Point Albino, Canada	1896	....	trace	.....	....	85.83	10.74	96.57	....	0.74	.....	2.69	945.8	953.2	0.6208
Vancouver, B. C.	.....	....	0.14	.....	....	.....	.....	93.56	....	.....	.....	6.30	.....	.....	.....
Bottineau, N. D.	....	3.00	.....	0.20	1.20	82.70	....	82.70	0.50	.....	.....	12.40	750.2	763.4	0.6246
Kenova, W. Va.	....	0.60	0.30	0.70	0.30	78.30	19.60	97.90	....	.....	.....	0.20	1026.2	1036.8	0.6621
Kenova, W. Va.	....	0.40	0.20	0.70	0.60	78.90	18.80	97.70	....	.....	.....	0.40	1019.5	1029.2	0.6551

found in the Kansas gases in percentages as high as 1.8, and in only one case of all the gases tested was none found. Inasmuch as no effort was made to detect this gas in any former analyses, it is probable that it was present, although undetected, since it was usual to consider as nitrogen all residue left after determining the other gases present.

20 From the above it will be seen that while natural gas may contain a considerable number of constituent gases, the great bulk of it is composed principally of but four or five of any importance.

21 In Table 1 the gases marked with an asterisk are samples that came from small shallow wells, or were otherwise abnormal, and thus cannot be considered as representative. Excluding these from the list and striking an average of those remaining, the constitution of what may be termed the average natural gas, after slight modifications in accordance with the above discussion, would be about as in Table 2.

TABLE 2 CONSTITUTION OF THE AVERAGE NATURAL GAS

Methane (CH_4)	87.00
Ethane (C_2H_6)	6.50
Ethylene (C_2H_4)	0.20
Carbon monoxide (CO)	0.20
Hydrogen (H_2)	trace
Nitrogen (N_2)	5.50
Carbon dioxide (CO_2)	0.50
Helium (He)	0.10
Oxygen (O_2)	trace

B.t.u. per cu. ft. at 29.82 in. and 60 deg. = 887.3.
Specific gravity = 0.6135.

22 *Heating Value of Natural Gas.* The fuel value of natural gas depends principally upon the relative proportions present of methane, ethane, and the inert gases, such as nitrogen and carbon dioxide. The only satisfactory manner of ascertaining the heating value is by means of calorimetric determinations, but as very few of these were reported with the analyses shown in Table 1, and in order to arrive at some comparative idea of the relative fuel values of the various gases listed, column 14 is given, showing the b.t.u. per cu. ft. as computed from the analyses. In making calculations, the constants in Table 3 were used.

23 In the discussion of properties of gases it is necessary to refer all volumes to some definite standards of pressure and tem-

perature. The standards most used in scientific work are 29.92 in. (760 mm.) of mercury, corresponding to 14.7 lb. per sq. in. for the pressure base, and 32 deg. fahr. (0 deg. cent.) for that of the temperature, while in industrial work 30 in. of mercury and 60 deg. fahr. are largely used. Some years ago, F. H. Oliphant, geologist for the Standard Oil Company, published a handbook on natural gas, in which he established the pressure base of 14.65 lb. per sq. in. absolute, and temperature base of 60 deg. fahr., and since that time it has been customary for natural gas men to refer their measurements to these bases. A pressure of 14.65 lb. per sq. in. is 4 oz. above an average assumed atmospheric pressure of 14.4 lb., the latter being the average at about the elevation of the Great Lakes, which elevation was considered as fairly representing that of most of

TABLE 3 HEATING VALUE AND SPECIFIC GRAVITY OF THE GASES FOUND IN NATURAL GAS

Kind of Gas	Symbol	Net Heating Value, B.T.U. per Cu. Ft.	Specific Gravity Air = 1.0
Methane	CH_4	897	0.5529
Ethane	C_2H_6	1594	1.0368
Ethylene	C_2H_4	1471	0.9676
Carbon monoxide	CO	322	0.9671
Hydrogen	H_2	271	0.0692
Hydrogen sulphide	H_2S	615	1.1769
Nitrogen	N_2	....	0.9701
Carbon dioxide	CO_2	....	1.5195
Helium	He	....	0.1382
Oxygen	O_2	....	1.1052

the gas fields. Thus, 4-oz. gas is the generally accepted pressure unit of measurement among natural gas men and will be so considered throughout this paper. Frequently gas is bought or sold on different pressure bases, and in such cases the corresponding absolute pressures are determined by considering atmospheric pressure as being equal to 14.4 lb. per sq. in., as for instance, a 10-oz. selling base corresponds to $14.4+0.625=15.025$ lb. absolute.

24 Conforming with this usage, therefore, all formulae, heating values, etc., throughout this paper are given for the unit of measurement as being 1 cu. ft. at 60 deg. fahr. and 14.65 lb. per sq. in. absolute (29.82 in. of mercury), which is commonly referred to as "gas standard."

25 It will be noted in Table 1 that the heating values given are

the net or low values. In publishing gas analyses or constants, many writers, in fact, most of them, give the high values, but the author believes the low to be more rational, inasmuch as in nearly all uses to which natural gas is put as a fuel, it is impossible to recover the latent heat of vaporization of the steam formed by the combustion, and therefore the net heating values give a more truthful conception of the actual worth of the gas for fuel purposes.

26 When it is impossible to obtain a calorimetric determination of the heating value of a particular gas, the next best procedure is to compute it from the chemical analysis of the gas, using the values shown in Table 3 for the heating values of the constituent gases. This, of course, is done by multiplying the percentage of each gas present by its corresponding heating value per cubic foot, and adding the products. The specific gravity is obtained by computation in precisely the same manner. Such computed results are necessarily subject to whatever errors there may be in the analysis of the gas, and unless this has been done with great care and precision, a wide discrepancy may exist between the calculated and actual values.

27 It is oftentimes desirable to gain an approximate knowledge of the heating value of a gas when neither a calorimetric determination nor a chemical analysis is available. In such cases, a fair "guess" may be made from a determination of the specific gravity of the gas, provided it is known to be a normal "dry" gas without freakish tendencies. The specific gravity is readily determined by the effusion method, in which the time required to pass a given quantity of the gas through a pin-hole orifice in a thin plate under a given head, or pressure, is divided by the time required to pass the same quantity of air through the same orifice and under the same pressure; the square of the quotient being the specific gravity of the gas, referred to air. For most reliable results the air and the gas should be run at the same temperature, to avoid the necessity of a correction for this factor.

28 An approximate formula for determining the heating value from the specific gravity may be derived from the following considerations. In the analysis given in Par. 21 for an average natural gas, and which represents the average constitution of the gases considered as representative in Table 1, it will be seen that of the combustible gases, methane and ethane comprise 93.5 per cent of the whole and ethylene and carbon monoxide comprise 0.2 per cent each. No great error will be made, therefore, if these two latter gases are con-

sidered as being a part of the paraffin group, especially since ethylene and ethane do not differ greatly in either heating value or specific gravity. The inert gases may likewise be combined in one group, of which the resultiug specific gravity may be considered equal to 1.0. Consequently, for approximate results, the average natural gas may be regarded as made up of three distinct gases, methane, ethane, and "inerts," of which the heating values and specific gravities may be considered as in Table 4.

29 Representing the relative volumetric proportions of these gases, expressed decimally, as m, e, and i, respectively, the following relations will obtain:

$$m+e+i=1.0 \quad [1]$$
$$H=897m+1594e \quad [2]$$
$$G=0.5529m+1.0368e+1.0i \quad [3]$$

in which H = the lower heating value in b.t.u. per cu. ft. of natural gas at gas standard, and G = the specific gravity of the gas. Eliminating m and e from equations [1], [2], and [3], the heating value of the gas may be expressed in terms of i and G as follows:

$$H=1440G-1541i+100.6 \quad [4]$$

30 The sum total of the percentages of the inerts given in Par. 21 is $0.061=i$. Substituting this in equation [4],

$$H=1440G+6.6 \quad [5]$$

Applying equation [5] to the average gas of Table 2, of which the

TABLE 4 HEATING VALUE AND SPECIFIC GRAVITY OF METHANE, ETHANE AND INERTS

Kind of Gas	B.T.U. per Cu. Ft.	Specific Gravity
Methane	897	0.5529
Ethane	1594	1.0368
Inerts	....	1.0000

specific gravity is 0.6135, it would indicate the gas to have a fuel value of $H=890$ b.t.u. per cu. ft., as compared with a value of 887.3 as computed from analysis.

31 The values of H and G given in Table 1, when plotted against each other, yield the result shown in Fig. 1. In this figure, only those gases considered as representative, and not marked with an asterisk, have been plotted. The resulting points are very widely scattered, but the general tendency is toward an increasing value of H with increasing G, as would naturally be expected, and it must be remembered that many of the analyses plotted cannot be vouched

for as to accuracy, in fact it is reasonably certain that they are inaccurate. The diagonal line drawn in the figure is the graph of equation [5], and is not far from an average line through the points shown. It is fair to conclude, therefore, that for purely approximate work, a reasonable notion of the heating value of natural gas may be obtained from the known specific gravity by using equation [5]. When the locality is known from which the gas is derived, the approximate proportion of the inert gases may be learned by selecting, from Table 1, a gas from the same field, and the heating value com-

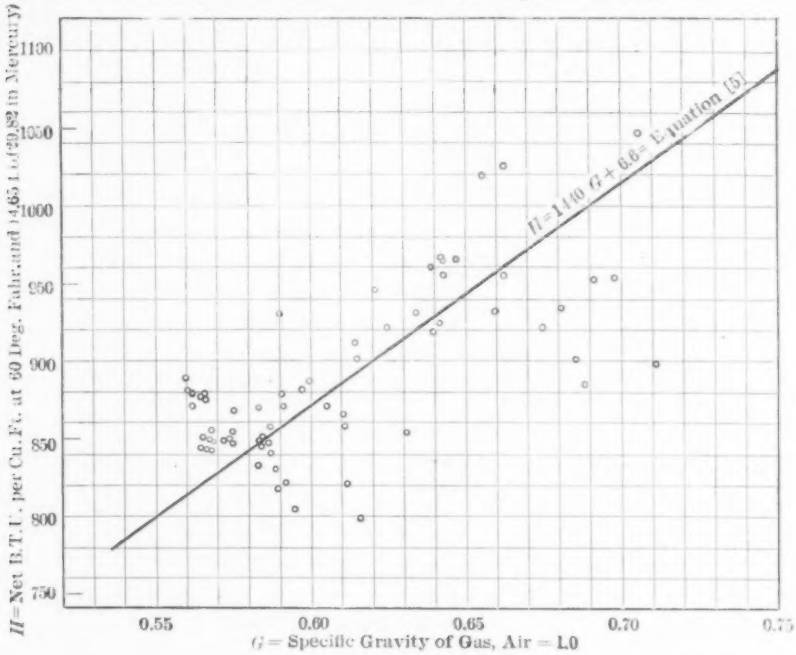

FIG. 1 CURVE SHOWING RELATIONS OF HEAT CONTENT AND SPECIFIC GRAVITY OF GAS, FROM TABLE 1

puted by means of equation [4]. The values calculated in this way are shown in column 15 of Table 1, and may be compared with those computed directly from the analyses.

32 As stated, no great degree of accuracy is claimed for this method of computation, but the author has frequently found it of great convenience where other and better methods were not available. The comparative values shown in the table as computed from the analyses and by equation [4], indicate that a fair degree of approximation can be attained by it.

33 Reference has been made to "dry" gas. This term is used to designate the ordinary natural gas coming from a well producing gas exclusively, as distinguished from that coming from an oil well, which is called "wet." These terms have no reference to water vapor in the gas, but imply the absence or presence of vapors of hydrocarbons higher in their respective series than those discussed above.

TRANSMISSION OF NATURAL GAS

34 *Pipe-Line Flow Formula.* In the design of pipe lines for the transmission of natural gas from the field to the points of consumption it is necessary to make use of a formula expressing the relations to each other of the quantity and initial and final pressures of the gas, and the diameter and length of line. Many such formulae have been proposed giving widely differing results. In nearly all of them the flow is stated as varying as the square root of the fifth power of the pipe diameter, and either the coefficient of friction is considered constant, or a different coefficient is given for each diameter of pipe. This serves well enough where the diameter is known and any one of the other quantities expressed by the formula is desired, but is somewhat awkward when it is desired to ascertain the diameter of line necessary to meet the other given conditions.

35 The author has derived a new formula, which he believes expresses the relationship of the quantities involved even more closely than any heretofore offered. It is based on isothermal flow, and the variation in the value of the coefficient of friction is provided for without complicating the formula, yet permitting the required diameter of line to be ascertained readily.

36 The expression for the initial velocity of any gas flowing in a pipe is given by Unwin [1] as

$$u_1 = \sqrt{\frac{gCTm}{fl}\,\frac{(P_1^2 - P_2^2)}{P_1^2}} \qquad [6]$$

where

u_1 = initial velocity, ft. per sec.

g = acceleration due to gravity

C = thermodynamic constant of the flowing gas $= \frac{PV}{T}$

T = absolute temperature of gas

m = hydraulic mean radius of the pipe $= \frac{D}{4}$

[1] Compressed Air, p. 67, Van Nostrand's Science Series.

P_1 = absolute initial pressure of the gas, lb. per sq. in.
P_2 = absolute final pressure of the gas, lb. per sq. in.
f = coefficient of friction
l = length of line, ft.

Let

C_a = thermodynamic constant for air
G = specific gravity of flowing gas, air = 1.0
D = diameter of pipe, ft.
d = diameter of pipe, in.

Then

$$C = \frac{C_a}{G}$$

and

$$m = \frac{D}{4} = \frac{d}{48}$$

Hence

$$u_1 = \left[\frac{gC_aT(P_1^2 - P_2^2)d}{48\,GflP_1^2}\right]^{\frac{1}{2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [7]$$

If

q = quantity of gas flowing per second, based on absolute pressure and temperature of P_o and T_o

A = area of cross-section of pipe in sq. ft. $= \dfrac{\pi d^2}{4\times 144}$

Then

$$q = u_1 A \frac{P_1 T_o}{P_o T} = u_1 \frac{\pi d^2}{4\times 144} \frac{T_o}{P_o} \frac{P_1}{T} = \frac{\pi}{576} \frac{T_o}{P_o} \left[\frac{gC_a(P_1^2 - P_2^2)d^5}{48\,GTfl}\right]^{\frac{1}{2}} \ldots [8]$$

If

Q = flow, cu. ft. per hr., based on P_o and T_o
L = length of line, miles.

Then

$$l = 5280L$$

and

$$Q = 3600q$$

$$Q = \frac{3600\pi}{576\sqrt{48\times 5280}} \frac{T_o}{P_o} \sqrt{gC_a} \left[\frac{(P_1^2 - P_2^2)d^5}{GTfL}\right]^{\frac{1}{2}} \quad \ldots\ldots\ldots [9]$$

Taking $g = 32.17$ and $C_a = 53.33$,

$$Q = 1.6156 \frac{T_o}{P_o} \left[\frac{(P_1^2 - P_2^2)d^5}{GTfL}\right]^{\frac{1}{2}} \quad \ldots\ldots\ldots\ldots\ldots [10]$$

37 Experiments on the flow of air in pipes of different diameters indicate that the coefficient of friction f is a variable, decreasing with increasing diameters of line. A great many such experiments have been collected and published in Compressed Air, by Elmo G. Harris, from which, by the use of equation [10], the coefficients of friction have been computed and plotted in Fig. 2.

38 In the reports of these tests no statements were made as to the method of measuring the quantity of gas flowing, and it is quite probable that many of the results are inaccurate in this respect. Notwithstanding this, however, the nature of the variation of f with

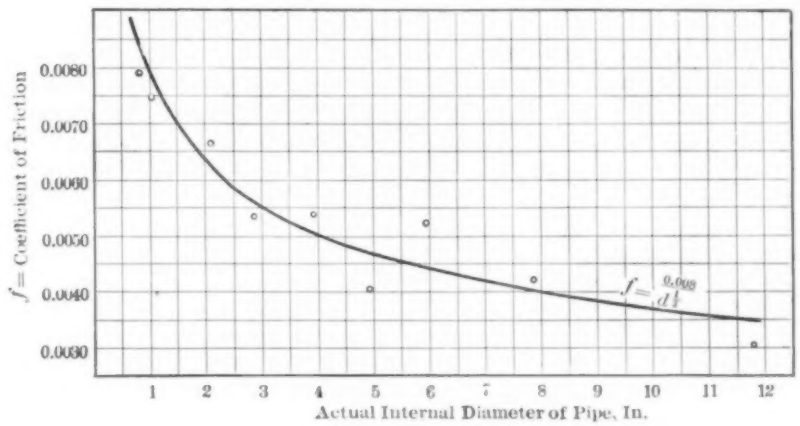

FIG. 2 CURVE SHOWING COEFFICIENT OF FRICTION OF FLOW OF AIR IN PIPES

the diameter is evident, and the curve represented by the equation

$$f = \frac{0.008}{\sqrt[4]{d}}$$

gives a fair average of the loci of the points plotted. Inserting this value of f in equation [10], the expression becomes

$$Q = 18.062 \frac{T_o}{P_o} \left[\frac{(P_1^2 - P_2^2) d^{5\frac{1}{4}}}{GTL} \right]^{\frac{1}{2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [11]$$

Equation [11] is the general formula for the flow of gas in long pipe lines.

39 In 1901, Forrest M. Towl conducted an extended test on an 8-in. line, 70 miles long, supplying gas to Buffalo, the results of which were published in a bulletin issued by Columbia University in 1911. Previous to the test the line had been repaired and tested for leaks, and was known to be practically gas tight. The flow was measured

by standardized pitot tubes, which gave results accurate within less than 1 per cent. The specific gravity G of the flowing gas was 0.64, its temperature, 32 deg. fahr., or $T=492$ deg. absolute. The temperature basis on which the gas was measured was 50 deg. fahr., or $T_o=510$ deg. absolute, and the pressure basis was 4 oz. above 14.4 lb., or $P_o=14.65$ lb. per sq. in. absolute. In a length of pipe 70.32 miles long, P_1 and P_2 were 210 and 41 lb. per sq. in. absolute, respectively. The actual diameter of the pipe was 7.981 in., and the rate of flow by pitot tube was found to be 221,000 cu. ft. per hr.

40 Inserting these quantities in formula [6] and solving for flow, it becomes

$$Q=221{,}400 \text{ cu. ft. per hr.}$$

or less than 0.2 of 1 per cent greater than the actual flow as measured.

41 Assuming gas standard conditions of measurement basis, namely, 60 deg. fahr. and 14.65 lb. absolute pressure, and that the average flowing temperature of the gas throughout the year will be 40 deg. fahr., the formula becomes,

$$Q=28.66\left[\frac{(P_1^2-P_2^2)d^{5\frac{1}{2}}}{LG}\right]^{\frac{1}{2}} \quad [12]$$

and, if an average specific gravity of 0.60 be assumed

$$Q=37\left[\frac{(P_1^2-P_2^2)d^{5\frac{1}{2}}}{L}\right]^{\frac{1}{2}} \quad [13]$$

42 Formula [13] is of practical use in designing lines for the transmission of natural gas. It is used as given, or in a transposed form, for all problems relating to single lines of uniform diameter.

43 If a line is composed of several lengths, L_1, L_2, $\ldots L_n$, of diameters d_1, d_2, $\ldots d_n$, each of these lengths must be transformed into an equivalent length of one chosen diameter, by means of the formula

$$L_n^1=L_n\left[\frac{d_n^1}{d_n}\right]^{5\frac{1}{2}} \quad [14]$$

These equivalent lengths added together will give $L=L_1'+L_2'+\ldots+L_n'$, which is the value for L in formula [13].

44 Values of equivalent lengths for different diameters can be most conveniently ascertained by the use of curves in Fig. 3, which consist of plots of the values of $d^{5\frac{1}{2}}$ for varying values of d. Values taken from these curves are convenient to use directly in the pipe-

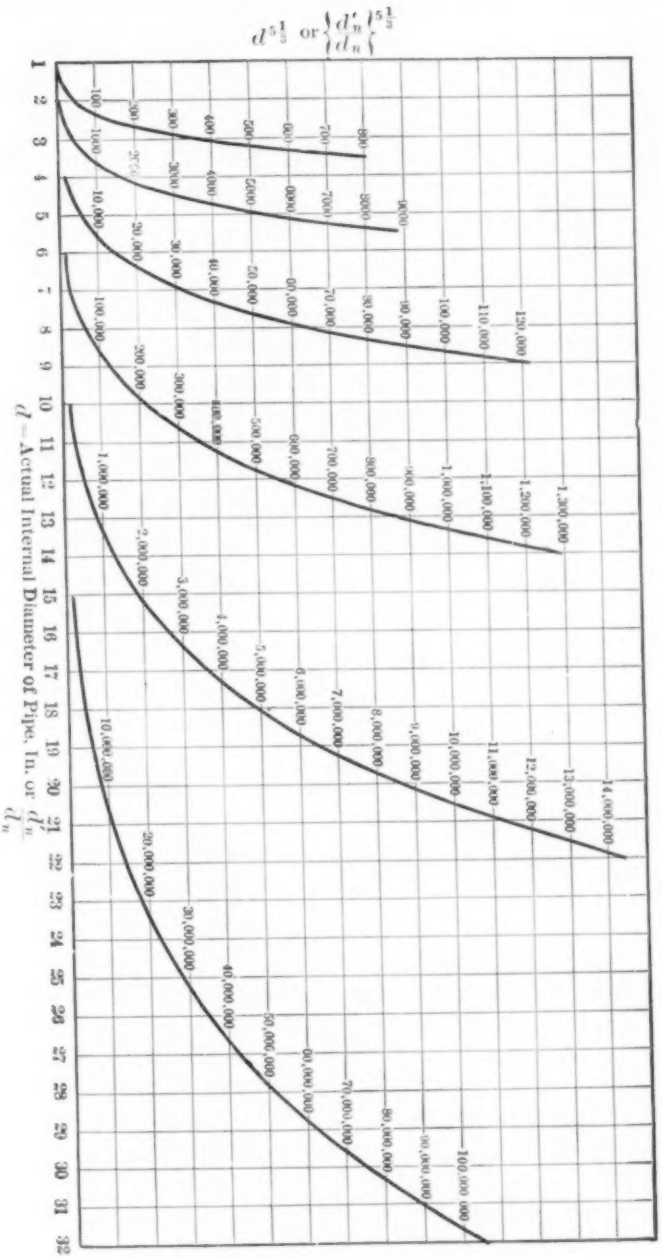

FIG. 3 CURVES SHOWING VALUES OF EQUIVALENT LENGTHS FOR DIFFERENT DIAMETERS OF PIPES

line formula, equation [13], whereas they are most simply used in equation [14] as values of the $5\frac{1}{3}$ power of the diameter ratios.

45 In the next case to be discussed, that of looped systems of lines (two or more pipes paralleling each other), it often happens that the several pipes of a loop have different lengths. It is then necessary, first, to transform all of them into their respective equivalents having a common length. This is done by means of a transposed form of equation [14], namely

$$d'_n = d_n \left[\frac{L'_n}{L_n}\right]^{\frac{3}{16}} \quad \ldots\ldots [15]$$

It will be seen, of course, that these values can likewise be derived by the use of Fig. 3.

46 Having reduced the pipes to their equivalents of a common length L, let the equivalent diameters be d_1, d_2, $\ldots d_n$. The initial and final pressures P_1 and P_2 will be common to all, and therefore the flow per hour through the system will be

$$Q = 37 \sqrt{\frac{P_1^2 - P_2^2}{L}} \left[\sqrt{d_1^{5\frac{1}{3}}} + \sqrt{d_2^{5\frac{1}{3}}} + \ldots + \sqrt{d_n^{5\frac{1}{3}}}\right] \quad \ldots\ldots [16]$$

$$= 37 \sqrt{\frac{(P_1^2 - P_2^2)}{L}} \Sigma \sqrt{d^{5\frac{1}{3}}} \quad \ldots\ldots [17]$$

$$= 37 \sqrt{\frac{(P_1^2 - P_2^2) d_o^{5\frac{1}{3}}}{L}} \quad \ldots\ldots [18]$$

where

$$\sqrt{d_o^{5\frac{1}{3}}} = d_o^{\frac{5}{2}} = \Sigma \sqrt{d^{5\frac{1}{3}}} = \Sigma d^{\frac{5}{2}} \quad \ldots\ldots [19]$$

or

$$d_o = \left[\Sigma d^{\frac{5}{2}}\right]^{\frac{2}{5}} \quad \ldots\ldots [20]$$

47 If the whole line consists of other sections of similarly composed looped lines, the equivalent single diameter of each section can be determined by equation [20] and the section then reduced to an equivalent length of some chosen uniform diameter by equation [14]. These equivalent lengths added together will give a line of uniform diameter of single pipe equivalent to the actual looped system.

48 For facilitating the computation of such a line, Fig. 4 is given, which is a plot of values of $\sqrt{d^{5\frac{1}{3}}}$ for values of d. From these curves

the values of $\sqrt{d^5}$ for the several loop diameters are read off and added together, the resulting equivalent diameter d_e being then ascertained from the curve.

49 *Pipe-Line Storage Capacity.* The storage capacity of a pipe line is the excess gas, over and above the average rate of supply, that

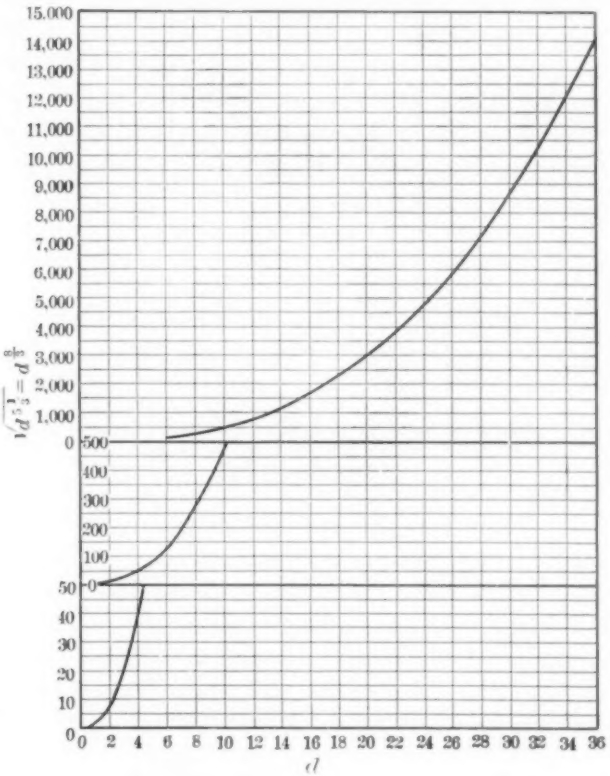

FIG. 4 CURVES GIVING DIAMETER OF PIPE RAISED TO $^8/_3$D POWER FOR USE IN COMPUTING PIPE LINES CONSISTING OF SECTIONS OF LOOPED LINES OF OTHER DIAMETERS

can be packed in the line during the period when the demand is less than the supply. The supply rate is the rate of average daily delivery and since the volume of gas contained in the line is a function of the pressure, it is necessary in order to determine the storage capacity, to develop an expression for the total line contents, which will take into account the variable pressure conditions at all points along the line. Having such an expression, the total contents under the

"packed" and "unpacked" conditions may be computed, and the storage ascertained by taking their difference.

50 The unpacked condition of a pipe line obtains when the gas is flowing at the average daily rate, for when the consumption is below this point all excess gas is being stored, and this point, therefore, is the lower limit, or base of the peaks in the daily delivery curve for the day.

51 The upper limit, or packed line condition, is not as readily determined, but if it is considered to be such that the intake pressure is at its maximum point, as fixed by considerations of safety, station pump pressure limits, etc., and the flow is a mean between the minimum and average rates for the day, the result will be very

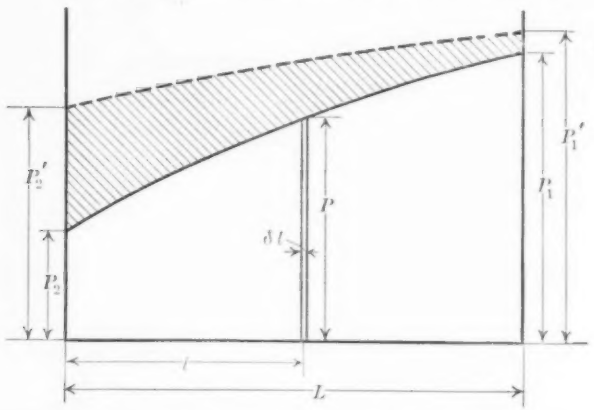

FIG. 5 VARIATION OF INTAKE PRESSURE P_1 FOR VARYING VALUES OF L, LENGTHS OF PIPE, WITH DISCHARGE PRESSURE P_2 CONSTANT

nearly actual conditions, and whatever error may thereby be involved will be on the side of safety in estimating the storage capacity.

52 Fig. 5 shows the variation of intake pressure P_1 for varying values of L with the discharge pressure P_2 constant. Two curves are plotted: The solid line represents the conditions with an assumed average rate of flow and the discharge pressure P_2 fixed by the minimum value allowable by the requirements of the distributing system. The dotted line represents the pressure conditions in the same line with the intake pressure P_1 at the maximum allowable value, and the flow at an assumed mean of the minimum and average rates for the day. This latter curve represents the packed line, and the shaded area the quantity of gas stored in the line available for peak loads.

53 Let P = absolute pressure at any point; then for any given line and flow condition, from equation [13]

$$P_1^2 - P_2^2 = l\frac{Q^2}{37^2 d^{5.3}} = Kl \quad \ldots\ldots [21]$$

wherein

$$K = \frac{Q^2}{37^2 d^{5.3}} \quad \ldots\ldots [22]$$

Hence

$$P = \sqrt{Kl + P_2^2} \quad \ldots\ldots [23]$$

The gas contained in a length δl based on any pressure P_o will be in cu. ft.,

$$\delta V = 5280\, A \cdot \delta l \cdot \frac{P}{P_o}$$

where A is the cross-sectional area of the pipe in sq. ft. The total quantity of gas in the line under the given conditions will then be

$$V = \int \delta V = 5280 \frac{A}{P_o} \int_o^L \sqrt{Kl + P_2^2}\, \delta l = \frac{3520}{K} \frac{A}{P_o} \left[(KL + P_2^2)^{\frac{3}{2}} - P_2^3 \right] \quad [24]$$

But

$$KL = P_1^2 - P_2^2 \text{ and } K = \frac{P_1^2 - P_2^2}{L}$$

Hence

$$V = \frac{3520\, AL}{P_o (P_1^2 - P_2^2)} \left[P_1^3 - P_2^3 \right]$$

$$= 3520 \frac{AL}{P_o} \left[P_1 + P_2 - \frac{P_1 P_2}{P_1 + P_2} \right] \quad \ldots\ldots [25]$$

$$= 19.20 \frac{d^2 L}{P_o} \left[P_1 + P_2 - \frac{P_1 P_2}{P_1 + P_2} \right] \quad \ldots\ldots [26]$$

54 In these formulae d is the actual internal diameter of the pipe in inches, and L the length of line in miles, the pressures all being lb. per sq. in. absolute.

55 In the case of a complex system it is necessary, first, to ascertain the common pressures at all junction points where the lines are tied together, and then to compute separately the capacity of each section. To do this it is necessary to use the actual diameter, or area, and length of each of the pipes in the loop. The gas content of a looped section thus becomes

$$V=\frac{3520}{P_o}\left[P_1+P_2-\frac{P_1P_2}{P_1+P_2}\right]\Sigma AL \quad \ldots\ldots\ldots\ldots [27]$$

or

$$V=\frac{19.20}{P_o}\left[P_1+P_2-\frac{P_1P_2}{P_1+P_2}\right]\Sigma d^2L \quad \ldots\ldots\ldots\ldots [28]$$

56 The gas content of the entire system will thus be the sum of the contents obtained for the several sections of line. If the quantity thus obtained for average flow conditions is deducted from that similarly computed for minimum flow with maximum intake pressure P_1, the result will be the total available storage capacity of the line.

57 If the letters used in equations [27] and [28] be taken to represent the pressure conditions of the packed lines and P_1' and P_2'

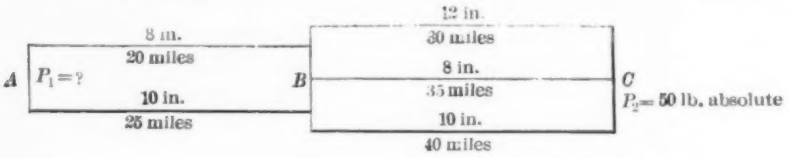

FIG. 5-a DIAGRAM ILLUSTRATING APPLICATION OF PIPE-LINE FORMULAE

those of the unpacked line, the available storage capacity of the system will be

$$S=\frac{3520}{P_o}\left(P_1-P_1'+P_2-P_2'-\frac{P_1P_2}{P_1+P_2}+\frac{P_1'P_2'}{P_1'+P_2'}\right)\Sigma AL \quad \ldots\ldots [29]$$

or

$$S=\frac{19.20}{P_o}\left(P_1-P_1'+P_2-P_2'-\frac{P_1P_2}{P_1+P_2}+\frac{P_1'P_2'}{P_1'+P_2'}\right)\Sigma d^2L \quad \ldots\ldots [30]$$

58 The following problem will illustrate the application of all of the above pipe-line formulae. Assume a complex line made up as shown in Fig. 5-a, and let it be required to find the equivalent line of single pipe and the intake, or initial pressure P_1 necessary to force 1,000,000 cu. ft. per hr. through the system, with a terminal pressure $P_2=50$ lb. absolute. Taking first the section from A to B, we have 20 miles of 8-in. (net) pipe and 25 miles of 10-in. By equation [15] the diameter of a pipe 25 miles long that will be equivalent to the 20 miles of 8-in. pipe is, in inches

$$d_n'=8\left(\frac{25}{20}\right)^{\frac{1}{16}}=8.342$$

Section AB is then equivalent to two parallel or looped lines each 25 miles long and having respective diameters of 8.342 in. and 10 in.

For $d = 8.342$ in.	$d^{\frac{8}{3}} = 286.2$ in.
for $d = 10.0$	$d^{\frac{8}{3}} = 464.2$
	$\Sigma d^{\frac{8}{3}} = 750.4$ in.

and by equation [20]

$$d_o = (\Sigma d^{\frac{8}{3}})^{\frac{3}{8}} = (750.4)^{\frac{3}{8}} = 11.97$$

or, section AB is equivalent to one pipe 11.97 in. in diameter and 25 miles long.

59 By equation [15] in section BC, the 8-in. pipe, 35 miles long is equivalent to a 7.772-in. pipe 30 miles long, since

$$d'_n = 8 \left[\frac{30}{35}\right]^{\frac{3}{16}} = 7.772$$

60 In the same manner, the 10-in. line 40 miles long, is equivalent to a 9.475-in. line 30 miles long. Then, by equation [20]

for $d = 12.0$	$d^{\frac{8}{3}} = 754.8$
$d = 7.772$	$d^{\frac{8}{3}} = 237.0$
$d = 9.475$	$d^{\frac{8}{3}} = 402.0$
	$\Sigma d^{\frac{8}{3}} = 1393.8$

$$d_0 = (\Sigma d^{\frac{8}{3}})^{\frac{3}{8}} = (1393.8)^{\frac{3}{8}} = 15.10 \text{ in.}$$

Hence section BC is equivalent to one pipe 15.10 in. in diameter, 30 miles long. By equation [14]

$$L'_n = 30 \left(\frac{11.97}{15.10}\right)^{5\frac{1}{3}} = 8.68 \text{ miles}$$

61 Thus, section BC is equivalent to a single line 11.97 in. in diameter and 8.68 miles long and the whole system is equivalent to a single line 11.97 in. in diameter and 33.68 miles long.

62 For ascertaining the required initial pressure for a flow of 1,000,000 cu. ft. per hr. the known line constants are therefore, $P_2 = 50$, $d = 11.97$, $L = 33.68$, $Q = 1{,}000{,}000$. By equation [13]

$$1{,}000{,}000 = 37 \sqrt{\frac{P_1^2 - 50^2}{33.68}} \overline{11.97^{5\frac{1}{3}}}$$

$$P_1 = 215.1 \text{ lb. absolute}$$

63 The relative quantities of gas passed by the several pipes of the loops are to each other as the $\frac{8}{3}$d power of their equivalent diam-

eters for equal lengths. The pressure at junction B can be ascertained by means of equation [13], using for L and d the equivalent values derived above for section AB. This pressure is required to be known in order to determine the storage capacity of the line. After having ascertained the above quantities, the pressure drops being known, the computation can be checked by computing the flow through the several lines as they actually exist.

64 Instead of computing the equivalent lengths and diameters by equations [14], [15], and [20], they may be ascertained from the curves of Figs. 3 and 4.

65 To compute the storage capacity of the system, first consider section AB. Assume the maximum allowable pressure at A to be $P_1 = 225$ lb. absolute, and the flow $Q = 800{,}000$ cu. ft. per hr. This section was shown to be equivalent to a single line 11.97 in. in diameter and 25 miles long. Substituting these known values in equation [13], the pressure at B is found to be $P_2 = 172.7$ lb.

66 When the line is unpacked; that is, under average flow conditions of 1,000,000 cu. ft. per hr. P_1' was shown to be 215.1 lb. and by formula [13], P_2' is found to be 117.4 lb.

67 Substituting these values in equation [30], together with the actual dimensions of the pipe in the section of line under consideration, the available storage capacity of this section AB in cu. ft. is

$$S = \frac{19.20}{14.65}\left(225 - 215.1 + 172.7 - 117.4 - \frac{225 \times 172.7}{225 + 172.7} + \frac{215.1 \times 117.4}{215.1 + 117.4}\right)(8^2 \times 20 + \overline{10}^2 \times 25) = 215{,}000$$

68 In like manner the storage capacity of BC is computed, the sum of the two results being the total storage capacity of the whole system available for peak demands.

69 *Power Required to Compress Natural Gas.* Before discussing the general problem of the design of a transmission system, it is necessary to give a rational formula for the power required to pump natural gas. It has generally been assumed that the compression is approximately adiabatic, and the theoretical adiabatic formula has been used for determining the power requirements. Many tests on compressors of widely differing types have convinced the author that this formula sometimes leads to serious errors, and, accordingly, an empirical formula is presented, which is believed to represent the actual operating conditions much more closely than the adiabatic

formula. First, however, both the isothermal and adiabatic formulae will be presented for purposes of comparison.

70 If gas could be compressed isothermally, the theoretical work required, as derived from the laws of perfect gases and with no clearance in the compressor, would be

$$W = P_1 V_1 \log_e \frac{P}{P_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots [31]$$

where

W = ft-lb. of work per lb. of gas compressed
P_1 = absolute suction pressure, lb. per sq. ft.
P_2 = absolute discharge pressure, lb. per sq. ft.
V_1 = volume of 1 lb. of gas at pressure P_1, cu. ft.

Let

P_o = absolute pressure, lb. per sq. ft. on which the measurement of the gas is based
T_o = absolute temperature basis of measurement, deg. fahr.
V_o = volume of 1 lb. of gas at P_o and T_o
T_1 = absolute temperature at which the gas is compressed

Then

$$\frac{P_1 V_1}{T_1} = \frac{P_o V_o}{T_o}$$

and

$$P_1 V_1 = P_o V_o \frac{T_1}{T_o}$$

hence

$$W = P_o V_o \frac{T_1}{T_o} \log_e \frac{P_2}{P_1} \text{ ft-lb} \quad \ldots\ldots\ldots\ldots\ldots\ldots [32]$$

71 At P_o and T_o 1,000,000 cu. ft. of gas will weigh $\frac{1,000,000}{V_o}$ lb. and if 1,000,000 cu. ft. are compressed in one day of 24 hours (1440 minutes), there will be $\frac{1,000,000}{1440\ V_o}$ lb. compressed per minute. Hence it will require

$$\frac{1,000,000}{1440\ V_o} \frac{P_o V_o T_1}{T_o} \log_e \frac{P_2}{P_1} \text{ ft-lb. per min.}$$

or

$$\frac{1,000,000}{1440 \times 33000} \frac{T_1}{T_o} P_o \log_e \frac{P_2}{P_1} \text{ h.p.}$$

72 If p_o = the base pressure in lb. per sq. in. absolute, and p_2 and p_1 are absolute pressures per sq. in. corresponding to P_2 and P_1

$$P_o = 144\ p_o$$

and

$$\text{h.p.}_i = \frac{1{,}000{,}000 \times 2.3026 \times 144}{1440 \times 33000} p_o \frac{T_1}{T_o} \log_{10} \frac{p_2}{p_1} = 6.978\ p_o \frac{T_1}{T_o} \log \frac{p_2}{p_1} \quad [33]$$

H.p.$_i$ is the theoretical isothermal horsepower per 1,000,000 cu. ft. of gas per day.

73 This formula gives the least possible power required to compress gas at the rate of 1,000,000 cu. ft. per day, and is therefore used as a measure of the efficiency of compression. In other words, "compression efficiency" is the figure obtained by dividing the h.p. as determined from equation [33] by that actually required on the compressor piston.

74 If the compression were performed strictly according to the adiabatic law where n is the ratio of specific heats of the gas at constant pressure and constant volume, the work, neglecting clearance, required for 1 lb. of gas would be, theoretically,

$$W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \quad [34]$$

But since

$$P_1 V_1 = P_o V_o \frac{T_1}{T_o}$$

$$W = \frac{n}{n-1} P_o V_o \frac{T_1}{T_o} \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \quad [35]$$

75 In order to compress 1,000,000 cu. ft. per day, it would require, as before,

$$\text{h.p.}_a = \frac{1{,}000{,}000}{1440 \times 33000 \times V_o} \frac{n}{n-1} 144\ p_o V_o \frac{T_1}{T_o} \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= 3.03\ p_o \frac{T_1}{T_o} \frac{n}{n-1} \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \text{h.p.} \quad [36]$$

76 As already stated, the bases generally adopted throughout the gas country are $p_o = 14.65$ lb. per sq. in. and $T_o = 60 + 460 = 520$ deg.; also, for average natural gas, $n = 1.266$. Substituting these values, and assuming $T_1 = T_o = 520$ deg., formula [33] for isothermal compression, becomes

$$\text{h.p.}_i = 102.23 \log \frac{p_2}{p_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots [37]$$

and formula [36] for adiabatic compression, becomes

$$\text{h.p.}_a = 211.27 \left[\left(\frac{p_2}{p_1} \right)^{0.29} - 1 \right] \quad \ldots\ldots\ldots\ldots\ldots\ldots [38]$$

77 Equations [37] and [38] give the theoretical h.p. per 1,000,000 cu. ft. per day, for isothermal and adiabatic compression respectively, on the assumption that the compressors have no clearance and that there are no losses of any kind. Obviously these conditions are impossible of realization in practice and experience shows that while one would expect the true formula to be somewhere between those for adiabatic and isothermal conditions, due to jacket cooling, in reality this is not the case, on account of the effects of mechanical clearance and slippage losses, to be explained later.

78 For isothermal compression the mean effective pressure in lb. per sq. in., as derived from equation [31], is

$$\text{m.e.p.}_i = 2.3026\ p_1 \log \frac{p_2}{p_1} \quad \ldots\ldots\ldots\ldots\ldots\ldots [39]$$

79 For adiabatic compression, derived from equation [34], it is

$$\text{m.e.p.}_a = \frac{n}{n-1}\ p_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \quad \ldots\ldots\ldots\ldots\ldots\ldots [40]$$

which, for natural gas, when $n = 1.266$, is

$$\text{m.e.p.}_a = 4.76\ p_1 \left[\left(\frac{p_2}{p_1} \right)^{0.21} - 1 \right] \quad \ldots\ldots\ldots\ldots\ldots\ldots [41]$$

80 If p_2 in equations [39] and [41] be regarded as constant and equal to 100 per cent, and p_1 be allowed to vary from 0 to 100, and the results plotted with values of m.e.p. as ordinates and values of p_1, as abscissae, the resulting curves will be shown as I and A respectively, of Fig. 6. The maximum, or peak load point for the isothermal curve I occurs at

$$p_1 = 0.3679\ p_2, \text{ or } \frac{p_2}{p_1} = e = 2.718$$

For the adiabatic curve A, it occurs at

$$p_1 = 0.3254\ p_2, \text{ or } \frac{p_2}{p_1} = n^{\frac{n}{n-1}} = 3.073$$

81 By referring to equations [37] and [38], it is seen that the power required to pump a given volume of gas is dependent solely upon the ratio of discharge to suction pressures, irrespective of the actual

value of either of them, whereas equations [39] and [41] show the mean effective pressure to depend upon the absolute value of the suction pressure, as well as upon the number of compressions. They show further that, as the suction increases from zero to the value of the discharge pressure, the latter remaining fixed, the mean effective pressure increases, theoretically, from zero to a maximum value, and then decreases to a theoretical zero when the suction is equal to the discharge. As a matter of fact, however, the indicated mean effective pressure in the compressor cylinder will not fall to zero with suction and discharge pressures equal, due to the wire-drawing of the dense gas through the valves, resulting in a diminution during the admission stroke, of the pressure in the cylinder below that in the suction line, and in a piling up, during the discharge stroke, of the pressure above that in the discharge line. Consequently, with suction and discharge pressures equal, the actual indicated power will always be greater than that of either the isothermal or the adiabatic.

82 With $p_1 = 0$, that is, when no gas is admitted to the compressor, the indicator card will show the compression and re-expansion lines to be practically superimposed if the valves are tight, and consequently the condition of zero work with zero suction pressure is very nearly attained in practice.

83 It has been the custom of most engineers to consider that the actual mean effective pressure of compression follows the adiabatic law, and they have therefore used equations [39] and [41], with constants depending upon assumed values of inefficiency, or lost work, according to the judgment and experience of the engineer. Repeated tests have shown, however, that the work curve has characteristics more nearly approaching those of the isothermal than those of the adiabatic, and in every case of which the author has any record the actual curve begins at zero, for zero suction, and rises between the isothermal and adiabatic curves until a point is reached, at about 3.2 compressions, where it crosses the adiabatic and thereafter continues above it. The peak load is found at about $p_1 = 0.42\, p_2$ or $\frac{p_2}{p_1} = 2.38$ compressions.

84 In Fig. 6 are plotted the results of two independent tests on compressors of large size. The circled dots represent the observations of a test reported by James S. Posgate, of the Kansas Natural Gas Company, in a paper[1] read before the Natural Gas Association

[1] Operation of Compression Stations, The Natural Gas Journal, August 1911, p 20

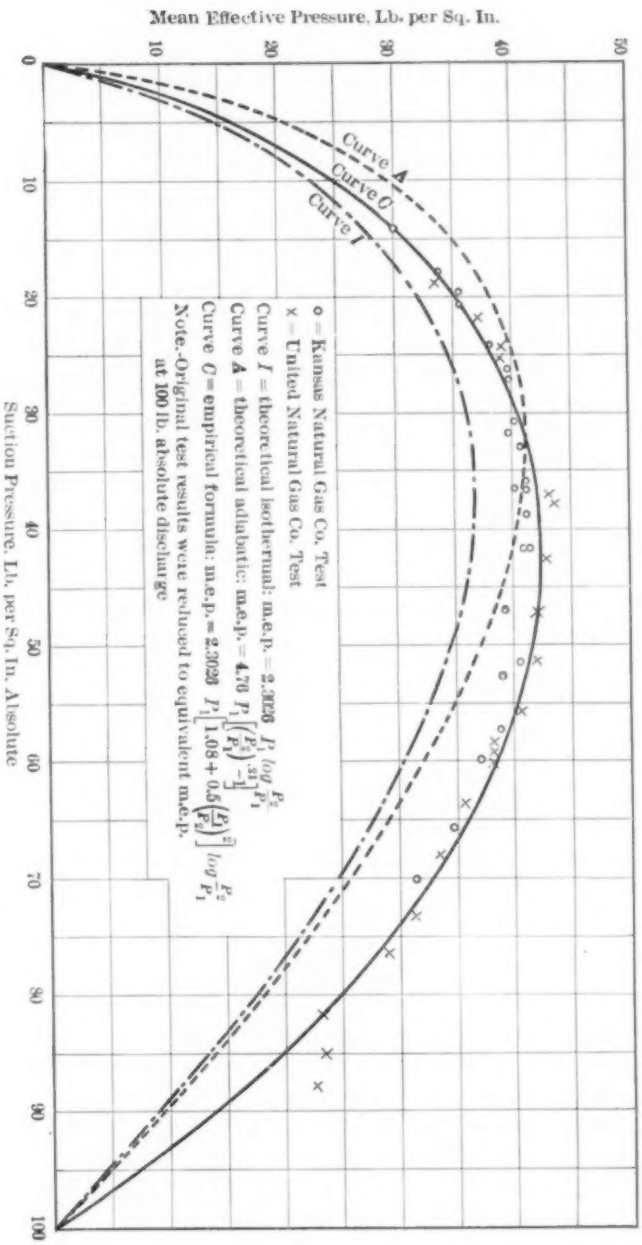
Mean Effective Pressure, Lb. per Sq. In.
0
10
20
30
40
50
60
70
80
90
100
Suction Pressure, Lb. per Sq. In. Absolute
Curve A
Curve C
Curve I
o = Kansas Natural Gas Co. Test
x = United Natural Gas Co. Test
Curve I = theoretical isothermal: m.e.p. = 2.3026 $P_1 \log \frac{P_2}{P_1}$
Curve A = theoretical adiabatic: m.e.p. = 4.70 $P_1 \left[\left(\frac{P_2}{P_1}\right)^{.21} - 1\right]$
Curve C = empirical formula: m.e.p. = 2.3026 $P_1 \left[1.08 + 0.5\left(\frac{P_2}{P_1}\right)^2\right] \log \frac{P_2}{P_1}$
Note.–Original test results were reduced to equivalent m.e.p. at 100 lb. absolute discharge

FIG. 6 RESULTS OF TWO INDEPENDENT TESTS FOR COMPRESSORS OF LARGE SIZE

of America at its sixth annual meeting, May 1911. The unit comprised two 17 in. by 48 in. compressor cylinders, operating at 260 lb. gage. The crossed points represent the observations of a test conducted by the author, in March 1911, on a $24\frac{3}{4}$ in. by 48 in. compressor running at 88 r.p.m. In the latter test the discharge pressure was held constant at about 103 lb. gage, and the suction pressure varied from 88 to 19 per cent of the discharge. The results of the observations in both tests were reduced to equivalent values corresponding to a constant discharge pressure of 100 lb. per sq. in. absolute.

85 A study of the plotted observations of these tests in comparison with the curves *A* and *I* in the same figure, reveals the fact that the actual curve attained in practice resembles the isothermal more

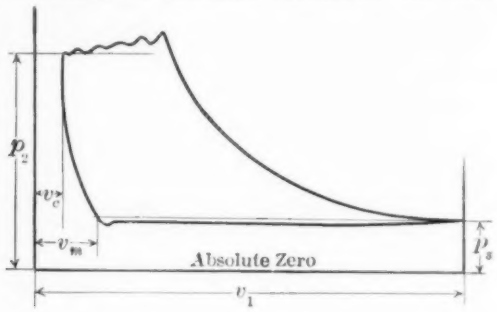

FIG. 7 TYPICAL COMPRESSOR DIAGRAM—SEE PAR. 89

nearly than it does the adiabatic; and that it diverges from the isothermal at a comparatively uniform rate as the suction pressure increases, and can be fairly represented by the equation

$$\text{m.e.p.} = 2.3 p_1 \left[1.08 + 0.5 \left(\frac{p_1}{p_2} \right)^2 \right] \log \frac{p_2}{p_1} \quad \ldots\ldots\ldots\ldots [42]$$

which is plotted as curve *C* in Fig. 6.

86 Curve *C* and its corresponding equation [42] may be taken as fairly representing the conditions met with in practice and has been found by the author to be safe to use in predetermining the indicated horsepower of a compressor.

87 The peculiar relationship existing between the actual and the isothermal curves is doubtless to be explained by the increased rate of cooling of the gas with increasing range of compressions, for the lower the suction pressure, and therefore the greater the number of compressions, the higher will be the temperature of compression and

the less the weight of gas pumped, with correspondingly less total quantity of heat generated. These results combined will therefore cause an increased rate of flow of heat into the cooling water, while at the same time less total quantity of heat will have to be extracted from the gas, with the result that the compression will more and more nearly approach isothermal conditions.

88 Inasmuch as the formula for the mean effective pressure represented by equation [42] is derived from actual operating conditions, it may be considered as taking into account all power losses. In the development of a working formula for the actual power required to pump a given quantity of gas, however, there are a number of sources of loss in volumetric efficiency which must be considered, the most important of which are as follows:

a Mechanical clearance
b Wire-drawing of the gas passing through suction valves and passages
c Heating of the inlet gas while entering the cylinder
d Valve and piston leakage

89 From a study of a great variety of indicator cards taken from many different types of compressors, the author has found that the re-expansion curve closely resembles the isothermal. In fact, many instances were found where its exponent was less than 1.0, doubtless due to discharge valve leakage. Assuming, therefore, that for practical purposes it may be considered as following the isothermal law,

$$p_2 v_e = p_s v_m$$

where p_2, p_s, v_e, and v_m are as represented in Fig. 7.
Hence

$$v_m = \frac{p_2}{p_s} v_e$$

Volumetric efficiency equals

$$E_i = \frac{v_1 - v_m}{v_1 - v_e} = \frac{v_1 - \frac{p_2}{p_s} v_e}{v_1 - v_e}$$

If the piston displacement = 1.0, then $v_1 - v_e = 1.0$ and $\frac{v_e}{v_1 - v_e} = v_e = m =$ clearance, expressed decimally. Hence

$$v_1 = 1 + m$$

and volumetric efficiency equals

$$E_i = 1 - m\left(\frac{p_2}{p_s} - 1\right) \ldots\ldots\ldots\ldots\ldots\ldots [43]$$

90 Equation [43] expresses the indicated volumetric efficiency of the compressor, i.e., the efficiency as measured on the indicator card.

91 The *indicated output* includes the effect of wire-drawing, or reduction of effective suction pressure of the gas due to passing through the suction valves and passages, as well as that of the indicated volumetric efficiency. In other words, the indicated output is determined by multiplying the piston displacement by the indicated volumetric efficiency and the absolute suction pressure *in the cylinder*, as measured from the cards, and dividing by the absolute pressure base on which the gas is measured. This wire-drawing effect reduces the compressor capacity and increases the power required to pump gas from a given line suction pressure in proportion to the extent of the pressure reduction. This is expressed as a percentage of the absolute line suction pressure. It must be included in the expression for indicated volumetric efficiency by making p_s in equation [43] equal to kp_1, where k is 1.0 minus the percentage drop in suction pressure due to wire-drawing, and p_1 is the line suction pressure, absolute. Hence, the indicated efficiency, expressed in terms of the line suction pressure, is

$$E_i = 1 - m\left(\frac{p_2}{kp_1} - 1\right) \quad [44]$$

92 The third source of loss, heating of the inlet gas entering the cylinder, is an effect that is difficult to determine directly. In 1909 the author conducted a series of tests on six different compressing units, three high-stage and three low, in an effort to discover to what extent this heating occurred, and found that, as was to be expected, it increased with increasing compression ratios, and was affected in a marked degree by any leakage that occurred through discharge valves or past the pistons. Its effect on volumetric efficiency was to reduce it in direct proportion to the ratio of absolute temperature in the suction line to that of the cylinder full of gas at the beginning of compression, and amounted roughly to about 1 per cent for each 5 deg. fahr. difference.

93 The fourth factor, namely, the loss due to leakage, reduced both the volumetric and the work efficiencies. While leaky suction valves caused a flattening of the compression line of the card and thus a reduction in the mean effective pressure, the quantity of gas pumped was also reduced with the net result of an increase in the power required for a given actual quantity of gas discharged. On the other hand, a leak back through the discharge valves into the

cylinder reduced the quantity of gas drawn in through the suction, not only by displacement, but also by its heating effect. It also raised the pressure during the compression stroke, making the compression line steeper, and thereby caused an increase in the mean effective pressure. The ultimate result, therefore, was an increased power requirement to pump a given quantity of gas.

94 In accounting for the losses due to heating and to leakage, they are usually combined with all unaccounted for losses, and called "slippage," for which a certain fixed percentage is assumed and inserted in the expression for volumetric efficiency.

95 Let s, expressed as a decimal, be the slippage effect. Then the actual efficiency or ratio of gas actually pumped, to that computed from piston displacement, would be

$$E = 1 - m\left(\frac{p_2}{kp_1} - 1\right) - s \quad \ldots\ldots\ldots\ldots\ldots\ldots [45]$$

and the quantity of gas pumped in one day of 1440 minutes would be, in millions of cubic feet, based on T_o and p_o

$$Q_m = \frac{1440}{1{,}000{,}000}\,\frac{A}{144}\,LN\,\frac{T_o}{T_1}\,\frac{kp_1}{p_o}\left[1 - m\left(\frac{p_2}{kp_1} - 1\right) - s\right] \quad \ldots\ldots [46]$$

The horsepower required on the compressor piston will be

$$\text{h.p.} = \frac{\text{m.e.p.} \times L \times A \times N}{33000}$$

The horsepower per 1,000,000 cu. ft. per day will thus be equal to

$$\text{h.p.}_m = \frac{\text{h.p.}}{Q_m} = \frac{100}{33}\,\frac{T_1}{T_o}\,\frac{\text{m.e.p.}}{\dfrac{kp_1}{p_o}\left[1 - m\left(\dfrac{p_2}{kp_1} - 1\right) - s\right]} \quad \ldots\ldots\ldots [47]$$

Substituting the value of m.e.p. as given in equation [42]

$$\text{h.p.}_m = \frac{100}{33} \times 2.3026 \times p_o \times \frac{T_1}{T_o} \times \frac{1.08 \times 0.5\left(\dfrac{p_1}{p_2}\right)^2}{k\left[1 - m\left(\dfrac{p_2}{kp_1} - 1\right) - s\right]} \log \frac{p_2}{p_1} \quad [48]$$

or, if $\dfrac{p_2}{p_1} = r$

$$\text{h.p.}_m = 6.978\; p_o\,\frac{T_1}{T_o}\,\frac{1.08 + \dfrac{1}{2r^2}}{k\left[1 - m\left(\dfrac{r}{k} - 1\right) - s\right]} \log r \quad \ldots\ldots [49]$$

96 For average conditions in practice, $m=0.02$, $k=0.98$, $s=0.04$, and $p_o=14.65$. Substituting these values in [49]

$$\text{h.p.}_m=102.2\ \frac{1.08+\frac{1}{2\,r^2}}{0.96-0.02\,r}\log r \qquad [50]$$

97 The efficiency of compression is equation [37] divided by equation [50], or

$$E_c=\frac{0.96-0.02\,r}{1.08+\frac{1}{2\,r^2}} \qquad [51]$$

98 In two-stage compression, for equal work in both stages, the power required is equal to twice that necessary to pump the gas through a single stage in which the number of compressions is equal to the square root of the total two-stage compression range. Therefore, if h.p.$_2$= the horsepower per 1,000,000 for two-stage operation,

$$\text{h.p.}_2=2\times 6.978\ p_o\ \frac{1.08+\frac{1}{2r}}{k\left[1-m\left(\frac{\sqrt{r}}{k}-1\right)-s\right]}\log\sqrt{r}$$

$$=6.978\ p_o\ \frac{1.08+\frac{1}{2\,r}}{k\left[1-m\left(\frac{\sqrt{r}}{k}-1\right)-s\right]}\log r \qquad [52]$$

Substituting the same constants as were assumed for equation [50]

$$\text{h.p.}_2=102.2\ \frac{1.08+\frac{1}{2\,r}}{0.96-0.02\sqrt{r}}\log r \qquad [53]$$

99 The simplest method of using the horsepower formula is by means of a plot giving values of horsepower per 1,000,000 cu. ft. of gas per day corresponding to values of $\frac{p_2}{p_1}=r$. Equations [37], [38] and [50] are so plotted in Fig. 8, together with the efficiency of compression, equation [51].

100 It will be seen that whereas the actual mean effective pressure in the compression of natural gas is less than that of true adiabatic compression for ranges above 3.2 compressions, the actual horsepower required to pump a given quantity of gas is always

greater, no matter what the compression range may be. This is due to the effect of clearance and the other losses discussed above.

101 The curves in Fig. 8 give the indicated horsepower required on the compressor piston, and in order to determine that necessary in the power cylinders, the values taken from the curves must be increased according to the mechanical efficiency of the machine.

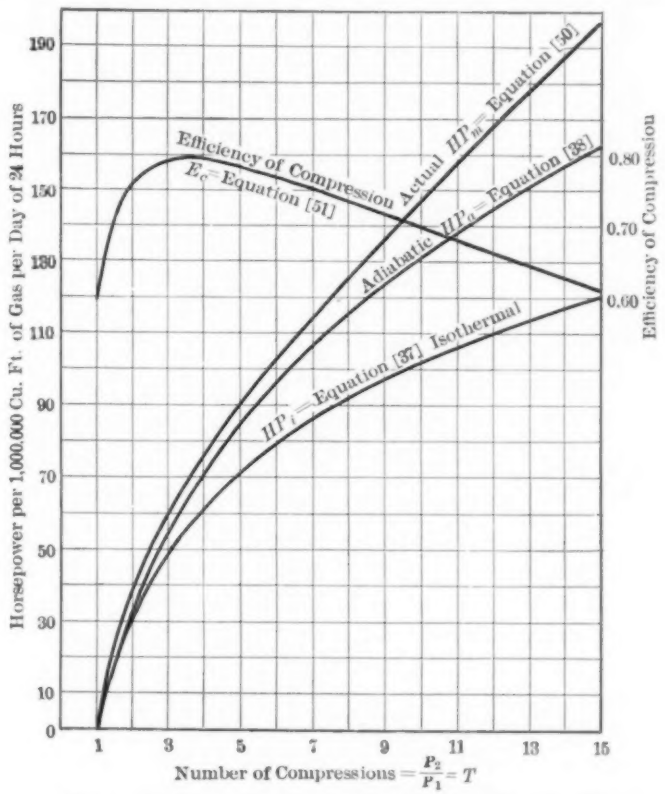

FIG. 8 PLOT OF EQUATIONS [37], [38], [50] AND [51]

This efficiency will range from 75 to 80 per cent in direct-connected gas-engine-driven compressors of large size.

102 Single-stage compression is advisable for ranges up to seven or eight compressions for the reason that the benefit that would be derived from two-stage operation is overbalanced by the excess first cost of the units, combined with the greater cost of operating, installing, and housing them. As the compression range increases above this point, however, two-stage operation becomes necessary

from an operating standpoint, and its advantage from an economical standpoint becomes more and more pronounced.

103 In compressing natural gas through two or more stages it is found to be impracticable to drive the compressors of all stages by the same power unit, on account of the variable pressure conditions which unbalance the loads of the different stages except at one particular ratio of compression. For this reason it is the usual practice to have the compressors of each stage driven by their own individual power units.

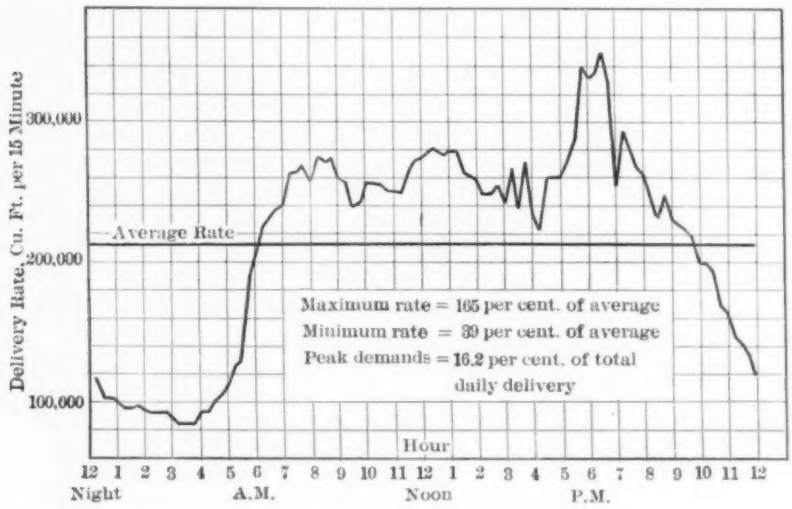
Delivery Rate, Cu. Ft. per 15 Minute
300,000
200,000
100,000
Average Rate
Maximum rate = 165 per cent. of average
Minimum rate = 39 per cent. of average
Peak demands = 16.2 per cent. of total daily delivery
Hour
12 1 2 3 4 5 6 7 8 9 10 11 12 1 2 3 4 5 6 7 8 9 10 11 12
Night
A.M.
Noon
P.M.

FIG. 9 LOAD CURVE SHOWING DELIVERY OF GAS DURING A COLD DAY IN WINTER

104 It is seldom, if ever, advisable to pump through more than two stages in natural gas transmission, for the reason that the pressure is limited by the strength of the pipe lines to a value that would not warrant the added complication and cost of multi-stage units higher than two.

105 *Pipe-Line Design.* In order to design a transmission system it is necessary to consider the character of service to be rendered and the various load factors to be expected. For domestic service in large cities the maximum daily delivery during the year will be about twice the daily average for the year, while the minimum will range from 25 to 30 per cent of the average. Thus, if 7,300,000,000 cu. ft. of gas are to be delivered in a year, or an average of 20,000,000 per day, provision will have to be made to transport 40,000,000 dur-

ing the coldest day of winter. The load curve for a typical cold day must then be considered and the pipe lines and stations designed not only to deliver the total quantity for the day at the average hourly rate, but also to provide sufficient storage capacity to carry the load safely over the peaks. A typical load curve for a cold day is shown in Fig. 9, and it will be seen that there are three main peaks, occurring about at meal times. As a rule, the demand for gas during the peaks will amount to about 16 per cent of the total daily delivery. The line storage capacity will therefore have to be equal, at least, to this amount. It is thus necessary to provide a line of sufficient capacity to transport the daily average with ease, and to have the requisite storage capacity. At the same time the pumping stations will be required not only to pump the average quantity of gas demanded, but will be required to pump it to a pressure sufficiently high to pack the line, and thus give it the proper storage. This packing must, of course, be done during periods of depression in the daily load curve, when the demand is less than the supply from the stations.

106 There are many variables involved in the proper design of a pipe line for greatest economy, such as: the pressure to be adopted; the size of the pipe with its limiting safe working strength; the number and location of pumping stations, and whether single or double stage, etc. All depend upon the conditions applying to the particular problem in hand. The fundamental principles upon which a line must be planned are those expressed by the pipe line and horsepower formulae already given, a fact which seems to the author to justify the space devoted to them.

107 In order to secure the greatest economy in design it is necessary to study the effects of the various factors involved in any given case, for there exists a certain relationship among pressures, pipe diameters and number of pumping stations that will give the maximum of economy for any given output and distance of transmission.

108 As a general rule, it is desirable to design the system for the maximum safe working pressures that the pipe will stand, for the reason that high pressures demand smaller pipes, and usually the pipe line is the heaviest item of cost of a system. High pressures, however, mean more power necessary in the compressor stations, with correspondingly increased cost of pumping the gas, thus partially, or perhaps completely, counteracting the lower cost of the smaller lines due to such high pressures. Line leakage is also much more serious the higher the pressure, and on this account it is neces-

sary to make the safe working pressure limits of the pipe line much lower than would be indicated by testing a single joint of the pipe alone.

109 The quantity of gas to be delivered also has a marked influence on the economical number of stations to be used, for the power necessary to pump it is directly proportional to the amount of gas pumped, and therefore the operating and first costs of stations likewise, whereas the size of the line, under the same pressure conditions, increases as the $\frac{3}{8}$th power of the quantity. Consequently, the greater the quantity, the greater will be the relative cost of power equipment over that of the line.

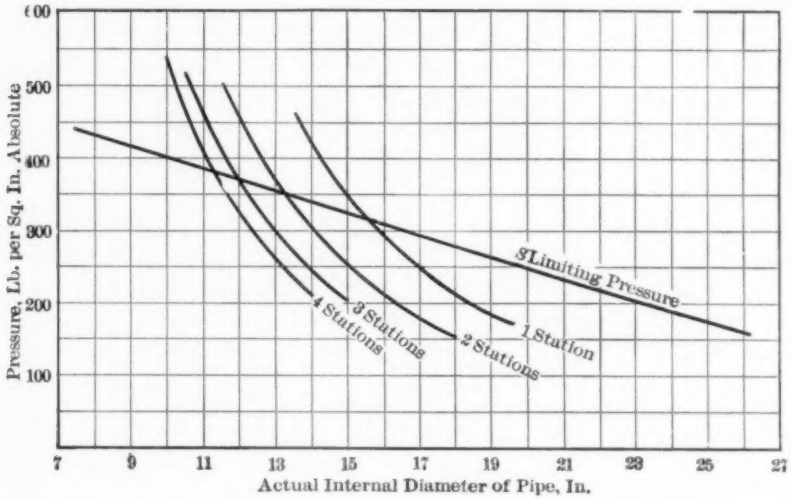

FIG. 10 PIPE DIAMETERS FOR INITIAL LINE PRESSURES FOR INSTALLATIONS OF FROM ONE TO FOUR STATIONS

110 Relay stations between the field and the delivery point should always be single stage only, for it requires the same amount of power to compress the gas through the first stage, say from atmosphere to 53 lb. gage; as it does through the second stage from 53 lb. to 300 lb., whereas practically no benefit is derived in the line for the expansion from 53 lb. to atmosphere, as compared with that from 300 to 53. This is seen from the pressure factor in the pipe-line formula.

111 In order to illustrate the effects on the problem of the various factors involved an example is worked out and plotted. It is assumed that the average daily delivery of a transmission system

during the coldest day is to be 24,000,000 cu. ft., or at a rate of 1,000,000 per hour. The transmission distance is to be 300 miles and it is required to determine the pressures and size of line as well as the number of pumping stations required. For the purposes of the example it is assumed that the total power installed will be equal to that determined by the formula without allowing for reserve capacity, and that the actual pipe diameters are used as derived, without regard to standard sizes. The cost of pipe line is taken as $900 per mile per inch of diameter, and the cost of compressor stations at $100 per compressor horsepower installed. Interest and depreciation on the line are taken at 10 per cent and on the power stations at 12 per cent, while operating costs are figured at $15 per year per installed compressor horsepower. The minimum allowable delivery pressure at the end of the line, where it feeds into the distributing system, is assumed as 50 lb. absolute, while the gas from the field is supposed to enter the compressors at 15 lb. absolute, or about atmosphere.

112 First, assuming one station in the field to pump the gas the whole distance, the required pipe diameters are figured for different initial line pressures by means of the pipe-line formula, equation [13]. These different values of pressure and diameter are then plotted as shown in curve 1, Fig. 10. In this figure the limiting safe pressures for various sizes of pipe line are also shown by the line S, and it is seen that this line and that of the required pressures cross each other. For any diameter less than that indicated by the intersection of line S with curve 1, the pressure would be too high for safe working and this point therefore indicates the minimum diameter and the maximum initial pressure that can be adopted with one station.

113 On the basis of the minimum diameter thus derived, the cost of the line is computed, and from the maximum pressure the power required to pump the gas is determined by means of the horsepower formula, using the proper value of suction pressure. From these computations the yearly cost is determined for a single-station equipment.

114 Then an intermediate station is assumed to be located midway of the distance. In computing the relative pressures and diameters it is first assumed that the intermediate station will pump through a single stage of 3.5 compressions, corresponding to the point of maximum compression efficiency as shown in Fig. 8. Under this assumption the terminal pressure of the first section of line is

$$P_2 = \frac{P_1}{3.5} = 0.2855 P_1 \quad \text{and} \quad P_1^2 - P_2^2 = 0.9184 P_1^2$$

With this value applied to the pipe-line formula the initial pressure for the field station is computed, assuming it to be the same for both stations. This causes the intermediate discharge pressure to be

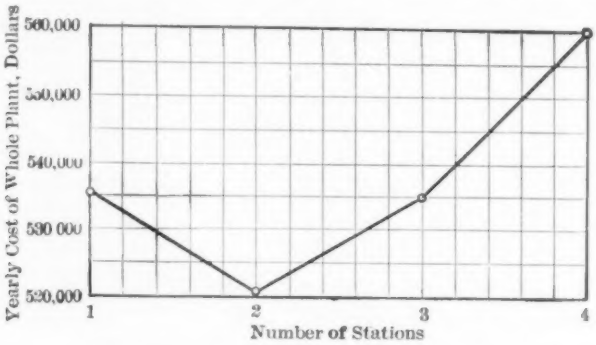

FIG. 11 CURVE SHOWING YEARLY COSTS AGAINST NUMBER OF STATIONS

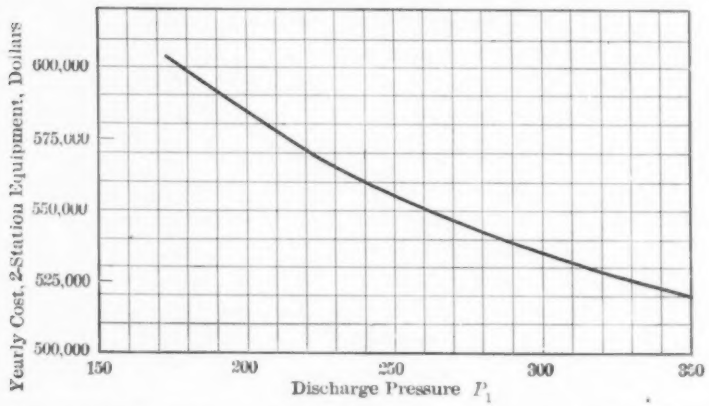

FIG. 12 CURVE SHOWING HIGHEST SAFE PRESSURES GIVING MAXIMUM ECONOMY

slightly higher than would actually be required to transmit the gas through the last section of line with its terminal pressure at 50 lb., but the increase is small, and the method insures the field station discharge pressure being at the proper value.

115 Values of pressure and diameter are plotted and the minimum diameter and the maximum pressure available are taken from the plot as before. The h.p. per 1,000,000 ft. of gas at the intermediate stations will then be constant at 70 h.p., corresponding to 3.5 com-

pressions, but the power of the field station must be figured for its proper ratio of compressions.

116 In the same manner the yearly cost is figured for three and four-station installations, and it is seen from Fig. 11, which is a plot of yearly costs against the number of stations, that for this particular case the lowest cost is attained with but two stations.

117 Figures made with all the conditions the same except that twice the quantity of gas is to be delivered, will show that a one-station equipment furnishes the most economical arrangement. Likewise with smaller maximum quantities to be delivered through the same distance, it can be shown that a greater number of stations, with correspondingly higher pressures and smaller line will result in the maximum economy.

118 Assuming the two-station layout and figuring yearly costs with decreasing initial line pressures, the curve in Fig. 12 results, showing that for this particular case the highest pressures safe to use give the maximum economy.

119 Having thus ascertained the most economical arrangement, it is necessary to consider the question of storage capacity, and see that when the line is packed the pressures and line sizes are adequate to provide sufficient storage gas to take care of the peak load.

120 If this is done for the example chosen it will be found that when the light deliveries are taking place at night the terminal pressure will back up to such an extent that with the intermediate station discharge pressure remaining constant the line will fill with gas to such an extent that more than sufficient storage will exist in the line. This illustrates one distinct advantage of large lines, especially at the terminal end of a system, where the gas can be stored at high pressure near the point of delivery and thus make it possible to draw on the reserve supply on short notice.

STATION DESIGN

121 In many natural gas fields, the natural pressure of the gas is sufficient to carry it to the market. The pressure declines, however, as the gas is withdrawn from the pools, and eventually a point is reached when it is necessary either to build more pipe lines to carry the gas, or to install pump stations for raising the pressure sufficiently high to make the existing lines serve. The principles already set forth will govern the choice in such a case. Usually, however, the pump station will be found preferable, for the reason that the assured continuance of the decline of pressure will eventually

result in the necessity for a station in any event, and, if installed in the beginning, it can readily be adapted to suit changing conditions.

122 Since the volume of gas increases as the pressure diminishes, the size of line necessary to convey a given quantity must increase likewise. Also the pressure drop becomes great under low-pressure conditions. In order, therefore, to make the work of the compressors effective in producing a pull on the wells without excessively low suction pressures at the station, it is necessary to have the stations located as near the center of the field as possible, thus reducing the length of suction lines to a minimum. A few miles added to the length of the discharge line mean almost nothing compared with their effect on the suction side. The location of compressor stations, therefore, means not only a study of the conditions relative to water supply and accessibility, but also of probable future developments in the gas production.

123 While passing, it is well to call attention to the necessity for the utmost care in making all suction lines absolutely tight to avoid the possibility of air entering them, and thus endangering the entire system, as well as impairing the service.

124 The type of compressor in general use for natural gas compression is the ordinary form of reciprocating air compressor. It must be equipped with automatic, or "voluntary," discharge valves, due to the fact that the expulsion of gas from the cylinder begins at differing periods of the stroke, depending on the compression ratio. The most satisfactory suction valve is likewise of this type, although mechanical suction valves have been tried with varying degrees of success. Because of the varying ratios of compression usually encountered, it is advisable to keep the clearance space in compressors as small as possible, for the higher the compression range, the lower will be the volumetric efficiency.

125 The design of the station units involves not only an assumption of the volume of gas to be pumped, but also the determination of the maximum discharge pressure to be demanded of the compressors.

126 The quantity of gas is usually estimated from a consideration of the expected demand for it, as well as a knowledge of the available supply. In most fields there are two general classes of wells, designated as "high pressure" and "low pressure," the former usually being the new ones with natural pressures sufficient to permit them to feed into the lines "ahead" of the pumps, that is, into their discharge lines. The latter, or low-pressure wells, are those that

require pumping. Frequently there are wells with pressures too low for feeding into the high-pressure discharge, but sufficiently high to feed the suction of the high-stage pumps, and by proper systems of piping, this is usually provided for. This naturally affects the station design, as it necessitates the pumping of more gas through the second stage than through the first, and proper provision must be made for it by the addition of the requisite number of high-stage units.

127 No fixed value of discharge or suction pressures obtains in the operation of any natural gas pumping station. The variation in these pressures throughout a single day may cover a wide range of limits, while it is especially marked from day to day throughout the year. In this respect the compression of natural gas differs widely from that of air under the usual conditions, and presents several additional problems not encountered in the compression of air. As relates to the question of design, this factor determines not only the power necessary to drive the compressor but also the quantity of free gas a machine will handle, and involves what is termed the peak-load condition of power requirement, as was shown in the development of the power formula. While it is well to provide for ample power to drive the compressors under all conditions of operation, the principle is easily carried to absurd limits, with the ultimate result of great loss in volumetric output. In some instances it is found necessary to pump gas with compression ranges varying from two to five or six. Under these conditions the unit must be designed to carry the load over the peak, and it will then operate under any and all conditions of suction pressure, with the discharge at its proper value. Usually, however, this will not be necessary, and then the upper limit of suction pressure may be estimated, and, if on occasion it should greatly exceed the value for which the units were designed, it can be reduced by throttling the gates on the suction lines.

128 The discharge pressure to be used in designing the compressor cylinders must be that of maximum condition, which is usually at night when the lines are being packed, or stored, with gas to provide sufficient storage capacity to help over the peak load conditions in the distributing system.

129 While this maximum pressure must be used to determine the engine power required, the ratio of cylinder sizes for two-stage compression must conform more nearly to average load conditions, in order properly to equalize the work between the two stages for average conditions of operation. Once the cylinder diameters are fixed, the ratio of discharge to suction pressure of the low-stage unit,

or, in other words, the ratio between the high and low-stage suction pressures, will be practically independent of the high-stage discharge pressure. The ratio of cylinder diameters may be determined as follows:

130 Let

V_l = the volume of low-pressure cylinder per revolution
D_l = diameter of cylinder
d_l = diameter of piston rod
s_l = stroke
n_l = speed, r.p.m.
e_l = volumetric efficiency
P_l = absolute suction pressure
T_l = absolute suction temperature

and let the corresponding letters, with sub h, apply to the high-stage compressor. Also, let P_o and T_o be the absolute pressure and temperature of any base on which the gas may be measured. Then the actual quantity of gas pumped per minute referred to P_o and T_o will be

$$Q = V_l \frac{P_l}{P_o} \frac{T_o}{T_l} e_l n_l = V_h \frac{P_h}{P_o} \frac{T_o}{T_h} e_h n_h \quad \ldots\ldots\ldots\ldots [54]$$

since all the gas that passes through the low-stage cylinder must be pumped through the high. Hence

$$n_l V_l e_l \frac{P_l}{T_l} = n_h V_h e_h \frac{P_h}{T_h}$$

$$\frac{P_h}{P_l} = \frac{V_l}{V_h} \frac{e_l}{e_h} \frac{T_h}{T_l} \frac{n_l}{n_h} \quad \ldots\ldots\ldots\ldots [55]$$

But

$$V_l = \frac{\pi s_l}{4} \left(D_l^2 - \frac{d_l^2}{2} \right) = \frac{\pi s_l}{8} (2 D_l^2 - d_l^2)$$

and

$$V_h = \frac{\pi s_h}{8} (2 D_h^2 - d_h^2)$$

Hence

$$\frac{P_h}{P_l} = \frac{s_l}{s_h} \frac{(2 D_l^2 - d_l^2)}{(2 D_h^2 - d_h^2)} \frac{e_l}{e_h} \frac{T_h}{T_l} \frac{n_l}{n_h} \quad \ldots\ldots\ldots\ldots [56]$$

Assuming the same speed and stroke for both stages,

$$\frac{P_h}{P_l} = \frac{(2 D_l^2 - d_l^2)}{(2 D_h^2 - d_h^2)} \frac{e_l}{e_h} \frac{T_h}{T_l} \quad \ldots\ldots\ldots\ldots [57]$$

131 From equation [57], it will be seen that the ratio of suction pressures in the two stages depends upon the net cylinder areas, the volumetric efficiencies and temperatures of the suction gas.

132 If the intercooling is inefficient, T_h will exceed T_l, and the discharge pressure of the low-stage compressor ($=P_h$=the high-stage suction) will be raised, thus imposing more work on the low-stage units. Usually, however, adequate cooling can be attained,

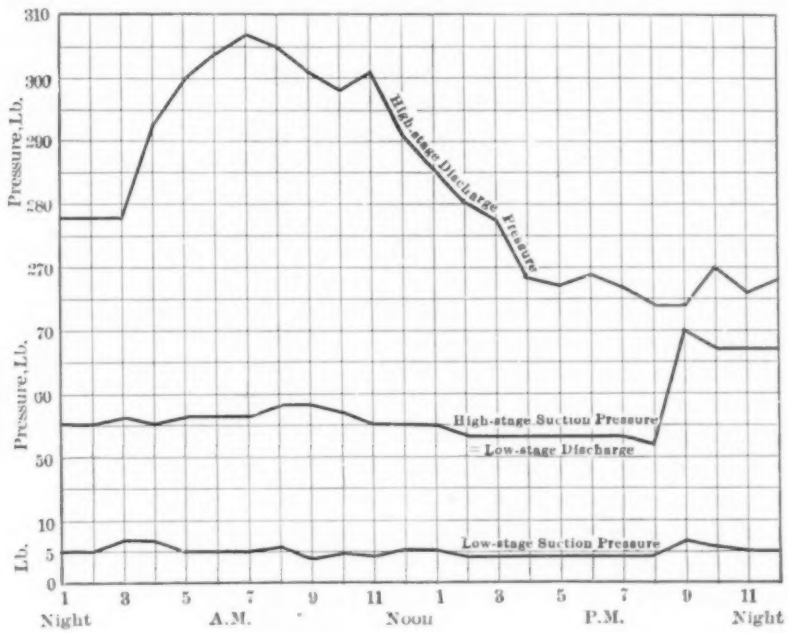

FIG. 13 CURVE SHOWING HOURLY PRESSURES OF TYPICAL PUMPING STATION FEEDING INTO A LINE 120 MILES LONG DURING AN AVERAGE WINTER'S DAY

and the ratios of temperatures may therefore be taken as one. Under this assumption the compression ratio in the low-stage cylinder becomes equal to the ratio of net effective areas of low to high-stage cylinders, that is, the computed areas multiplied by the respective volumetric efficiencies.

133 The ratio of the diameters of the high and low-stage cylinders can therefore be determined from equation [57], having fixed upon the proper ratio of compressions for each stage, as determined by average operating conditions.

134 In order to illustrate the fact previously stated, that the high-stage suction pressure is practically independent of its dis-

charge, Fig. 13 is given, showing a plot of the hourly pressures of a typical pumping station feeding into a line 120 miles long during an average winter's day.

135 The size of compressor cylinder required depends upon three things: the quantity of gas to be pumped, the suction pressure at which it can be supplied to the compressor, and the speed of the compressor.

136 The first of these requirements must be ascertained from the number of units to be installed, and, in this particular, there seems to be a considerable difference of opinion among engineers. The author believes that there should be a sufficient number of units, so that in case of a shutdown of any one of them, the capacity of the plant would not be seriously impaired. In fact, it is distinctly the part of wisdom to provide one spare unit over and above the actual requirement under the most severe conditions. Natural gas compressors, unlike most other power apparatus, cannot have their capacity forced, for there are only two means of increasing it, namely, by increasing the speed, which has a well defined upper limit, and by raising the suction pressure; and usually, when the greatest demand is made on the plant, the field is supplying all the gas of which it is capable, and thus it is impossible to add to this pressure. Natural gas companies are, as a rule, public service corporations, and it is absolutely necessary to maintain the operation of the stations continuously during severe weather in order to prevent great suffering.

137 Having given the quantity of gas to be pumped and the suction pressure, the required size of the cylinder is readily computed when the proper value of volumetric efficiency is assigned.

138 In good modern compressors of large size and long stroke, the indicated volumetric efficiency may range from 98 per cent to 90 per cent, depending upon the ratio of compressions, and the slippage loss will range from 3 per cent to 10 per cent, so that it is not unusual to find actual volumetric efficiencies as high as 94 per cent. In the majority of cases, however, the actual efficiency will range from 80 to 90 per cent, with 85 as a fair average for working conditions.

139 The constantly growing demand for compressors of high efficiency for natural gas compression has given a considerable impetus to improvements in design, which have resulted in lower clearance spaces and improved valves and jacket cooling, and it is entirely reasonable to specify volumetric efficiencies of 90 per cent, or even higher, with units of large size.

140 When the actual volumetric efficiency of a unit is known, its pumping capacity can conveniently be determined from the formula:

$$Q = 0.000909 \times E \times s \times \left(D^2 - \frac{d^2}{2}\right) \frac{NPT_o}{P_o T} \quad \ldots\ldots\ldots\ldots [58]$$

where

Q = output, cu. ft. per min.
E = volumetric efficiency of compressor
s = stroke, in.
D = compressor cylinder diameter, in.
d = compressor piston diameter, in.
N = revolutions per minute
P = suction pressure, lb. per sq. in. absolute
T = suction temperature, absolute
P_o = pressure base of gas measurement, lb. per sq. in. absolute
T_o = temperature base, absolute

141 This formula assumes that the compressor has no tail rod. Should it have one, the value in the parenthesis would have to be modified to suit. From the formula the diameter necessary for any given capacity can be computed by substituting the known quantities in the equation.

142 Usually the temperature of the suction gas is assumed to be the same as that of the measurement basis in stating the capacity of a compressor, and hence the factor $\frac{T_o}{T}$ in the formula is dropped, being equal to 1.

GENERAL REMARKS

143 The problem of the number of compressions to use in the determination of the power required is usually more or less indeterminate and requires the exercise of the judgment of the engineer in each particular case, based upon a knowledge of the field pressure conditions and the probable discharge pressure as already discussed. The discharge pressure is, as shown, a matter of computation, but the suction pressure may vary widely under different conditions.

144 Compressors for natural gas were originally driven by steam engines, but, owing to the recent improvements in the design and reliability of large gas engines, and to their high economy, this type of prime mover is rapidly supplanting the steam engine. Usually

the compressor is direct-connected to the engine, although there are many successful installations using the belt drive. Motor-driven units are rare however.

145 The tendency in recent years has been to lay gas transmission lines with rubber-packed sleeve couplings, in sizes above 4 in. In fact, it is imperative with lines above 10 in. in diameter, owing to the difficulty of making tight screw-collar joints with the larger sizes of pipe. Lines are usually designed for pressures as high as the pipe will stand, sometimes exceeding 400 lb. per sq. in., and, inasmuch as the leakage is a function of the pressure, irrespective of the quantity of gas flowing through the line, it is essential that the pipe joints be made as tight as possible.

146 In using the sleeve type of coupling it must be remembered that it offers very little resistance to end thrust, and proper provision must be made for anchoring the pipe wherever it deviates from a straight line, such as making turns or going over the crest of a knoll.

147 Heavy fittings are usually necessary, not only to stand the gas pressures, but also to provide against the enormous strains produced by expansion and contraction due to temperature changes and to frost. This is especially true at compressor stations because of the high temperatures due to compression, which sometimes reach 300 deg. to 350 deg. fahr. At these stations it is usual to provide swing-joint connections; all stiff joints should be avoided, especially where any considerable length of pipe is involved.

148 Owing to the possibility of leakage gas accumulating in a pump station and endangering the plant, it is necessary to provide adequate ventilation. The method favored by the author is to have the pump house equipped with a monitor, and a fan installed in the auxiliary building, which draws its supply of fresh air from outside, over the hot gas engine exhaust pipes which are laid in a conduit in the pump house. The air is then forced back into the pump house, thus providing both heat and ventilation without the necessity of auxiliary boilers or fires about the premises.

149 All buildings must be lighted by electricity, the generator being installed in the auxiliary building. This generator is usually of the direct-current type in order to provide current for charging the storage batteries generally installed for ignition purposes. The auxiliary building should be from 30 to 40 ft. away from the main pump house in order to lessen the risk of explosion.

150 All electric lamp sockets in the main pump house should be

of the keyless type, and all electric switches should be enclosed. Care should be taken to exclude any pockets under the pump-house floor or elsewhere where gas might collect.

151 In piping up gas compressor cylinders it is strongly advisable to provide a by-pass for use in starting, especially with gas engine driven units; and also an automatic safety valve, connected to the discharge pipe between the compressor cylinder and the first gate. There have been several instances where compressors have been destroyed by attempting to start with a closed discharge gate, and this precaution is most necessary. Such relief valves should have ample capacity to pass all the gas the compressor might discharge, and should be tested at regular intervals to insure their being in an operative condition at all times. They are fully as important as a safety valve on a boiler.

152 Two-stage pumping stations should be provided with both inter-coolers and after-coolers, which are usually composed of networks of pipe immersed in a pond or reservoir adjoining the station. Drips should be connected at the discharge end of all coolers, in order to collect and dispose of the gasolene which usually is produced by the process of compressing and cooling natural gas.

153 These drips are sometimes provided on an elaborate scale, where it is found that the gasolene production is sufficiently large to warrant collecting and selling it, and in some cases refining plants are installed for rendering the gasolene fit for the market.

154 It is important to extract all of this gasolene from the lines, as otherwise it will collect in depressions and "freeze," completely plugging the line. The secret of getting out the gasolene is so to build the after-cooler as to cause the flow of gas to be greatly retarded, when the liquid may be extracted by proper separators.

155 By-pass connections should be provided around the coolers in order to enable the plant to discharge its gas direct into the lines in case of leakage in the coolers.

156 All station piping should be "flexible," so as to permit operation with various systems of connection, and should be arranged in such a manner that the plant can be tested with ease.

THE EFFICIENCY OF CONTROL SYSTEMS IN FLYING MACHINES

At the meeting of the Society held in Boston on February 16, 1912, a paper was presented by Albert A. Merrill[1] on The Efficiency of Control Systems in Flying Machines.[2]

The first question to be considered is the movement of the levers connected with the control surfaces. The movement of these levers should be instinctive, that is, they should be such as one would make without being taught. To lower the front of the machine the lever should be moved forward, to raise the front it should be moved backward. This is done in all machines and there is never any danger of making a false movement with the fore and aft control. When the machine tips laterally, the lever should be moved towards the high side as, when tipping over, it is instinctive to move towards the high side.

In many machines this method is used, but in some, notably in the Wright machine, lateral control is obtained by a fore and aft movement of the lever. It can not be considered the best system, however, since it is not an instinctive motion and when placed in a dangerous position a man might not make the correct movement. The Wrights use this system because it requires considerable pressure to warp the wings of their machine and more pressure can be delivered by the arm with a fore and aft motion than with a lateral motion. However, the work of the Farman and Curtiss machines shows that ailerons will maintain lateral stability as well as warping wings, and as it takes but little pressure to move ailerons a lateral motion will give all the pressure needed.

Let us consider the aerodynamic efficiency of control surfaces in reference to the lift and drift of these surfaces. By lift is meant that component of the normal pressure at right angles to the line of flight, and by drift, that component parallel to the line of flight.

[1] Brookline, Mass.

[2] Published in abstract only. Complete report may be consulted in the rooms of the Society.

If two curves are drawn representing the lift and drift of a plane surface at all angles from 0 to 90 deg., it will be found that the lift component increases rapidly from 0 to about 20 deg., slowly from that angle to about 35 deg. and then drops rapidly to nothing at 90 deg. The drift component on the other hand increases constantly from 0 to 90 deg. These two curves show that the maximum pressure that can be given by a surface is its drift at 90 deg. Moreover, only at small angles can its lift be utilized efficiently, since with an increase of the angle, the drift increases, and this drift tends to reduce the total lift by reducing the speed. *It follows therefore that all control surfaces should be placed where their drift can be utilized.*

The first practical application of this is to the rudder. As now placed at the rear, only the lift component is used for turning, the drift component acting as a drag and retarding the speed of the machine and consuming power. The place for the rudder is not at the rear, but at the tips of the wings, because when placed at the tips, both components are utilized. In this position, the drift component turns the machine and the lift component tends to prevent skidding, thus reducing the required angle of banking. Rudders thus placed will satisfy the requirement that control surfaces should be placed where their drift can be utilized.

The three common systems now in use violate the requirement regarding control surfaces as just stated. The Wright system consists of means for increasing the positive angle of the wing to be raised, decreasing the positive angle of the wing to be lowered, and turning the rudder towards that wing, having the lesser angle of incidence. The Farman system consists in increasing the positive angle of an aileron on the wing to be raised and turning the rudder towards the other wing. The Curtiss system consists in increasing the positive angle of an aileron on the wing to be raised and turning not the rudder, but another aileron on the other wing to a negative angle so as to produce the necessary offsetting backward pressure.

The point common to all is that the positive angle of the wing or aileron, which it is desired to raise is increased and a backward pressure on that wing or aileron thus introduced. This backward pressure of course reduces the speed of the affected side, and unless the speed of the other side is reduced equally, the side desired to be raised will fall. It follows that in these systems the control surfaces are placed in such a position that the drift can not be utilized but must be offset, which offset is made either with a rudder or another

aileron. It is well to remember that pressures vary as the square of the speed and that a change of angle has little effect on lift compared to the effect of a change of speed. Thus if the angle is doubled, say a change from 4 to 8 deg., an increase of lift of only 27 per cent will result; but if the speed is doubled, four times the lift is obtained. It follows therefore that the difference in speed between the wing tips is of more importance than the difference in the angles of incidence. Broadly speaking, the three systems referred to are similar and all three are inefficient in so far as they introduce a force, drift, which can not be used but must be offset.

The error lies in attempting to raise the wing. It is impossible to do this without introducing drift, which antagonizes the lift component. When a flying machine loses its lateral stability nothing should be done to the low side, all attention should be devoted to the best way of introducing a downward and backward pressure on the high side. This can be done in a way which is exactly the reverse of the Farman system: Have the ailerons movable only to a negative angle and only on one side at a time, then always move the aileron on the high side. This is the simplest possible system, both mechanically and aerodynamically; the lift component will be downward and will lower the wing directly, while the drift component can be used because it comes on the high side and will retard, and hence lower, that side. The drift is used, no offset is needed, and moreover, there is no danger from over-control because, no matter how large a drift is introduced, from too great a movement of the aileron, it always helps to produce the desired result. This is not true of the drift component of the other systems. The writer feels sure that in time a system of control based upon this principle will drive out all other systems, for it is essential to stability systems that no force which can not be used shall be introduced.

Turning now to the question of fore and aft control and starting with the fundamental lift and drift curves, if a horizontal line is drawn at that point where the drift equals the thrust, and a new lift curve computed, assuming the thrust and hence the drift constant, but the angle increasing it will be seen that the lift drops off very rapidly with every increase in the angle of incidence. From this new curve three angles show important data: the minimum flying angle, which is the lowest angle at which the machine will fly; the climbing angle, which is the angle at which the machine will climb most rapidly; and the stalling angle, which is the maximum angle at which the machine will fly. Safe flying means that the angle of incidence must

be kept between the minimum and the stalling angles and since with any but the highest powered machines the difference between these two angles is small there is constant danger of stalling and having an accident. For this reason one ought always to have a good clinometer on the machine so as to know at a glance his flying angle. Except for this danger of stalling, fore and aft stability seems to be easier to maintain than lateral stability, partly because when lateral stability is lost the machine may start sliding sideways and this introduces very grave dangers.

Many inventors have proposed to use vertical surfaces to prevent this sliding, others have proposed the use of the dihedral angle, but to all such proposals there is a fundamental objection, which is that such systems do not begin to work until the machine has already begun to slide, whereas a control system should anticipate the action it is designed to prevent.

To sum up then the three important points regarding control systems: first the lever should be moved towards that part of the machine it is desired to lower; secondly all control surfaces should be placed if possible where the drift component can be utilized; and thirdly a control system should anticipate the action it is desired to prevent.

FLOUR MILLING AND MILL ENGINEERING

At the meeting of the Society held in Philadelphia, March 30, 1912, a paper was presented by B. W. Dedrick, instructor of milling and mill engineering, Pennsylvania State College, State College, Pa., on Flour Milling and Mill Engineering.[1]

The ancient builder of mills was called the millwright, and, even up to comparatively modern times, was the only person capable of designing, building, and placing machinery and planning buildings for their accommodation, not only of flour mills, but factories or whatever machinery was employed.

The modern flour mill building does not depart radically from other buildings designed for the reception and operation of heavy machinery, but the height of building and floors is given more consideration. Mill buildings are usually carried up three, four, five or more stories above the basement. It is necessary and better economy to have sufficient height in order to place certain machines one above the other, so as to take advantage of the force of gravity in the handling of stock. In designing mill buildings, the engineer must also take into consideration the great weight to be supported, not only of the heavy and numerous machines required and their driving connectors, spouts and bins, but of the grain and flour that is kept within the building. Besides, in the modern mill nearly all machines used are rapidly revolving, and if improperly placed or balanced, are particularly hard on a building, causing a perceptible weaving motion or tremor. The floors have to be of proper height to give sufficient pitch or incline to spouts, for if the spout is too flat or of insufficient pitch, the flour or other material will not run down, but choke or clog. The flour mill building, especially on the grinding and bolting floors, should be well lighted by a sufficient number of windows in day time, and electric or other good light at night, for it is necessary frequently to inspect the flour and other stock, and a good steady and bright light is absolutely essential to detect the extremely close shades in the color of flour, and compare it with the standard samples.

[1] Published in abstract only. Complete report may be consulted in the rooms of the Society.

Practically the first step of the wheat in the mill is in the separators, of which there are usually two: the receiving, that rough cleans the grain as it comes into the mill, taking out straws, wheat heads, sticks, oats, seeds and other trash; and the milling separator, that handles the wheat, cleaned almost entirely of foreign matter, making a closer and finer separation of the wheat, taking out any remaining oats, white caps, joints and seeds left over by the receiving separator. The light and imperfect wheat is drawn out by air current. From the separators the thoroughly cleaned wheat goes to the scourer which has a cylinder or series of blades or beaters revolving very rapidly, from 500 to 700 revolutions. These beaters throw the grain against the "case" which may be simply of heavy woven wire, the mesh or openings being small enough to prevent even the smallest berry from passing through, but allowing pieces of bran, chit or seed to be forced through, to settle in the hopper below, or to be drawn into the fan.

The effect of the beaters throwing the wheat against the rough casing at varying angles is to scour off the beards, exterior dirt and, in case of severe scouring, the outside cuticle or woody covering, all this being forced through the openings of the case, with the beaters acting as blowing fans. In both cases of the separator and scourer, the wheat is subjected to a current of air or suction at the receiving end and again on leaving the machine; this takes out the lighter wheat and remaining dust. On their passage to the fan, the heavier grains are arrested when they get into the tips where the expanding air loses some of its force, and cannot carry them farther, so they fall of their own weight into what is called "the screening tips." In some mills wheat that contains a great deal of cockle seed is passed through a cockle-cylinder which removes seed more perfectly. Mills that use wheat containing a good deal of smut or contaminated by smut, wash the wheat in machines for that purpose, after which it is dried and again scoured and tempered before going to the rolls.

A modern roll called a "double stand" consists of two pairs, one on each side. If each pair handles different material as they do in smaller mills, a partition below the rolls practically makes each a distinct pair.

The rolls are carried on bearings, which are rigid for one roll and movable or free for the other, allowing this roll to be adjusted closer to or further apart from the other roll according to the fineness desired to break or crush the stock passing between the rolls.

The arms of the movable roll are pivoted below on eccentric bearings, one on each side of the roll frame. The tramming or bringing the rolls into level is accomplished by means of turning this eccentric bolt, and then locked by a nut, but the arm is allowed to move or swing freely as in adjusting the rolls or spreading them apart, by the lever, when shutting down or when the roll runs empty. The arm at the top carries a recess which contains a heavy coil spring, which bears against the inner closed end, and against the wheel at the outer end. The rod passing through is the adjusting or lighter, corresponding to that of the millstone. At the outer end is the hand wheel used to compress the spring and thus stiffen or increase the tension. The inner hand wheel is the adjustor. By turning the wheel left or right, or forward on the threaded rod, it permits the rolls to close up; turning it in the opposite direction, spreads the rolls apart. Passing through the center under the feeder housing is a small shaft to which is attached a lever. There are two pairs of eccentrics on each end, to which are attached the adjusting rods by their yokes. The movement of the lever in one direction throws both the movable rolls from their mates simultaneously at both ends, and vice versa, brings them back to position.

Each pair of rolls has a fast and slow roll. In order to grind or crush with effect, there must be more or less of a rending or tearing action, and for this reason the rolls are given "differentials" in speed, the most common differentials used being $2\frac{1}{2}$ to 1 for break rolls on wheat, to $1\frac{1}{2}$ to $1\frac{1}{4}$ to 1 on the middling reduction rolls, which are smooth almost without exception, though very finely corrugated, or scratch rolls are sometimes used on the first clean middlings.

There are several types of bolters or sifters. The hexagon or six-sided reel is the oldest type of bolter still used, but its action in bolting is rather harsh. The round reel dresser is gentler in action, and like the hexagon, is slow in running, 30 revolutions being the mean. The centrifugal reel is used only on very soft, flaky material, for its disintegrating effect. The gyrating type of sifter is altogether considered to be the best and most gentle of all. The action of the gyrating sieves is to compel the stock to travel spirally on the surface of the flat sieve and on top of the silk, so as to bring the lighter, fluffier and impure parts on top of the traveling stream and hold them there until they tail off, the purer going through the cloth. Various means or devices are used to keep the meshes of the cloth open.

From there the wheat goes to purifiers. In the old flat grinding process all the flour possible was made at one and the first grinding. This system was very simple, consisting of a rolling screen, a smutter, a pair of burrs, one or two long reels, and a couple of elevators. Some middling or semolinas were made, but of such a character and containing so much fine bran or germs that when these middlings were reground on stone and bolted, it made an inferior flour as compared with the first, or superfine, produced in the grind of the wheat itself. The invention of a successful middlings purifier changed all that. From seeking to grind low, making no middlings or as little as possible, it now became the object to grind higher, cut the bran as little, and make as much middlings as possible. The miller reduced the feed on the stone by half, bestowed more skill and patience in dressing the burr, giving smoother surfaces and grinding higher and cooler with a millstone; he thus produced middlings that after being purified and then ground separately on stones and bolted, made flour superior to the "first" or "clear" flour produced from the grinding of the wheat itself. This flour produced from the purified middlings was called "New Process" or "Patent" flour.

The different grades of middlings, after being purified, are crushed on the smooth rolls and then sent to different sections of the sifter and sometimes to reels, as in the last roll reduction. With each reduction and subsequent separation of bolting, there is a certain amount of flour sifted out, and as a natural consequence there is an increased proportion of impurities to be handled on the next reduction and separation, so that the tendency is for the flour to be less white and clear, and partially to counteract this tendency it is necessary to have the cloths finer and finer. After certain reductions the tailings may be of such a nature as not to require further handling, and are sent direct to feed as finished, while those containing any floury parts are again reduced and sifted, the final results being flour, bran and shorts.

The paper contained also a detailed discussion of the capacity of machines and weights, and a description of the model mill to be built at Pennsylvania State College, State College, Pa., to give students the practical part of milling, and for experimental purposes in the way of testing the different kinds of wheat, to ascertain the bread making qualities of the flour; baking and chemical tests of flour and grain, and experimentation with various machines as to efficiency, power absorbed, etc.

DISCUSSION

In the discussion which followed, Mr. Wilson W. Welsh gave some reminiscences of Philadelphia milling in the early sixties, including a list of mills in existence at that time. Philadelphia of today has lost its former prestige as a milling center. From the early sixties the West was growing and built new mills, and was adopting all the latest improvements, while the eastern mills held back. The result was that theWest gained control in the eastern markets, and not until within the last twenty-five years did the Eastern millers wake up to the fact that they must keep pace with these improvements in order to retain their markets or draw even a small share of the business. The failure to adopt the improvements as they came forward was the cause of the decrease of milling in the East, not only in Philadelphia, but throughout the states of Pennsylvania, New York and Maryland. Now the situation has changed, the Eastern millers are alive to the necessity of adopting improvements, and are gaining ground.

Prof. J. P. Jackson drew a parallel between the development of the study of metallurgy and that of the production of breadstuffs. Five or six years ago it seemed as if we had reached the limit in the production of iron and steel in the quality of material, but the study of metallurgy went ahead, and today it looks as though we are only at the starting point. In the study of the production of foodstuffs this country has fallen behind Germany, but is going to be first again. It seems necessary and desirable to make some center where there would be a radiating influence or source of information from which an advantageous study of milling processes could be carried on, and where a young man could be prepared to go into the mills to work and advance their efficiencies. With regard to industrial training generally, the speaker wanted to see an opportunity for every boy, who was not going to take up professional work, to have a chance to use his hands as well as his brain.

Mr. H. V. White[1] told the story of how he and the Pennsylvania Millers' Association had attempted to secure from the state legislature a grant of funds for putting up the model mill described by the speaker of the evening, and emphasized the importance of such a mill. No one scarcely knows what scientific knowledge is necessary to drive a mill. A man who goes into a large mill today to learn milling is specialized and works in a groove, and may work for a dozen or more years without becoming a miller; he learns but one side of that work, and only few get down to the mechanics of milling.

[1] President, Pennsylvania Millers' Association.

FOREIGN REVIEW

BRIEF ABSTRACTS OF CURRENT ARTICLES IN FOREIGN PERIODICALS

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FOREIGN REVIEW

The aim of the Foreign Review is to present, within the available space, the main data contained in the articles indexed. Where possible, reference is made to English or American publications containing fuller information on the subject treated. Measures are given both in original units and their English equivalents. In many instances, engravings and tables are reproduced. Opinions expressed are those of the reviewer, not of the Society. Artic les are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *t* theoretical. Articles of exceptional merit are rated *A* by the reviewer.

Aeronautics

WINDKRÄFTE AN EBENEN UND GEWÖLBTEN PLATTEN. DRAHTWIDERSTAND. O. Föppl. *Zeit. für Flugtechnik und Motorluftschiffahrt*, March 16, 1912. 3½ pp., 10 figs. *eA*. Data of tests on the *action of wind on flat and curved surfaces and resistance of wires* made in the Laboratory for Testing Models at Göttingen, and comparison of these data with those obtained by other experimenters.

MOTORE "EOLO." ANNALI DELLA SOCIETA DEGLI INGEGNERI ED ARCHITETTI, October 16, 1911 (quoted from *Annales des Ponts et Chaussées*, January February, 1912). *d*. This *motor*, invented by Landriani and Carnaghi, *for use in aviation*, consists essentially of a cylinder *C* (Fig. 1) turning around a transversal axis *A* and provided with two pistons *P* enclosing a chamber *S* of variable volume; each piston has a piston rod terminating with a guide-pulley *R* running along a closed curve with center on the transversal axis *A*. If an explosion occur in the chamber *S* when the cylinder is in one of its intermediary positions, the tangential components of the pressures transmitted to the guide pulleys will form a couple which will cause the cylinder to rotate around the axis *A*. If the area of the curve be divided into four quarters, as shown in the figure, the total cycle will consist of admission of the mixture in quarter I, compression in II, explosion in III, and exhaust in IV.

SUR LE GYROPTÈRE, Papin and Rouilly. *Comptes rendus des séances de l'Academie des Sciences*, March 4, 1912. 1½ pp. *d*. Exposition of principles of construction and description of a new type of aircraft. A *gyroptcr* is a single-screw helicopter in which the fulcrum for the formation of the motor couple is provided not by a second screw as has been usually done, but simply by the resistance of the surrounding air, by means of

compressed air jets, these jets acting on the propeller screw in the same way as steam jets in a wind valve. The principle of the apparatus, and particularly that of the construction of the screw, *has been borrowed from the seed-carrying leaf of the sycamore*, the screw (Fig. 2) being single-bladed, and having its center of gyration practically independent of the speed according to general laws governing the equilibrium of non-symmetrical bodies rotating in space, such as lassos and boomerangs. In this apparatus the center of gyration is located in the car, which is enclosed in the body of the apparatus, supported and guided by it, but at the same time free to preserve its relative immobility, or to have its position con-

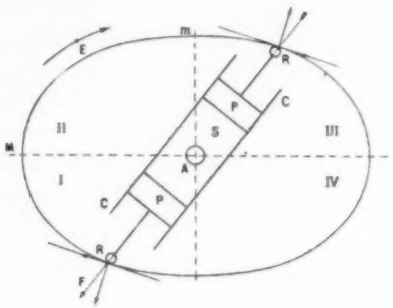


FIG. 1 "EOLO" AVIATION MOTOR

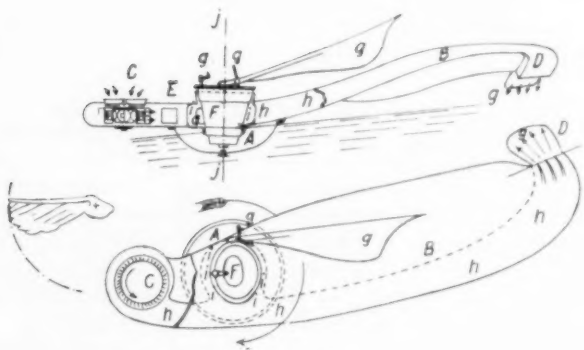


FIG. 2 GYROPTER OR SINGLE-SCREW HELICOPTER (CONSTRUCTED ON THE PRINCIPLE OF A SYCAMORE LEAF)

trolled by the aviator. Should the motor suddenly stop, the gyropter, owing to its distribution of weight and convenient choice of angles of planes, will automatically take a position such as to produce the slowest possible rate of fall as is done by the sycamore leaf. To rise, a convenient inclination must be produced, while the horizontal progression requires another inclination of the axis of the screw, which acts thus either as an

elevating, or as a driving propeller, or in both capacities at once. The article does not state whether the gyropter has been tested in actual flight, or what the size of the model constructed was.

HYDRAULISCHE HÖHENSTEUERUNG UND STABILISIERUNG DER FLUGZEUGE. Donát Bánki. *Zeits. für Flugtechnik und Motorluftschiffahrt*, March 16, 1912. 3½ pp., 3 figs. *d.* Description of a new *hydraulic stabilizer for aircraft.* The author started with the idea that all stabilizers based on the use of what he calls geodetic direction must be failures because the action of masses produces oscillations which have to be suppressed at the expense of sensitiveness of the apparatus. In Bánki's system of stabilization the whole control is located in the steering wheel, the airship being turned right and left by moving the wheel within the horizontal plane as in an automobile, while by changing the plane of the wheel itself the up and down motion of the ship is controlled: to do this, a system of servomotors is installed, keeping the plane of the ship always parallel to the plane of the steering wheel. There was, however, another difficulty to overcome. When the airship has been tilted by the action, say, of the air current, and the servomotor begins to act to set it straight, the opposition produced by it to the tilting action of the air current will grow until that action is overcome. It will not, however, stop right then, but continue to act, and, as a result, will tend to tilt the motor the other way. Then the stabilizer will begin to act in the other direction, and the process will be repeated, the aircraft meanwhile being subject to more or less violent oscillations. The essential part of Bánki's construction for bringing back the ship into its original position is a lever of which the "three fulcra" are connected with the adjusting gear of the governor, the distributing valve, and the cylinder of the servomotor. These three fulcra, or points, may be designated by *a*, *b* and *c*. When the ship tilts, the governor gear displaces *a*, and causes the lever to rotate about *c*, thus bringing the point connected with the distributing valve out of its middle position. This causes the motion of the cylinder of the servomotor and point *c*, and thus rotation of the lever about the point *a*. But while *a* is in motion, *b* cannot attain its middle position, because it is all the time kept out of it by the rotation about *c* as an instantaneous center of rotation. But at the instant when the governor gear is at rest, i.e., when there is no tilting of the ship, the lever begins to rotate about the instantaneously fixed point *a*, the distributing valve comes into its middle position, and the stabilizer ceases to act until it is brought into action by a new tilting of the aircraft.

Fig. 3 shows the construction of the Bánki stabilizer. The servomotor piston *1* is stationary, the liquid being brought in and out through the piston rods 2 and 3. The cylinder 4 moves up and down, is provided with a rack 5 engaging with a toothed segment 6, and thus driving a shaft on which is set lever 16 or a rope pulley. The shafts belonging to the two servomotors, one for longitudinal, and the other for lateral stability, are placed concentrically. The distributing valve is 8, sitting in the distributor case 7, and actuated on by the lever 14. The second fulcrum of this latter is on the cylinder of the servomotor, and the third is connected with the

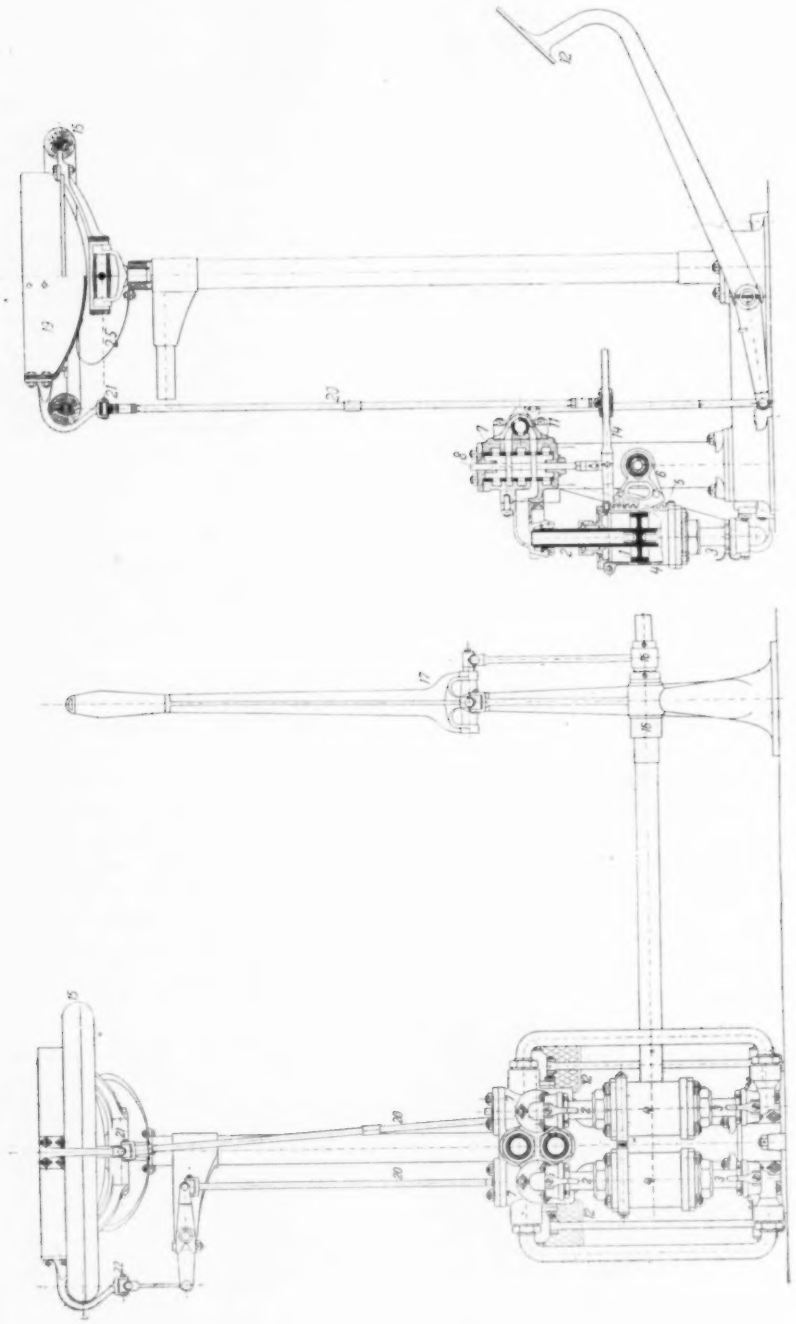


FIG. 3 BÁNKI STABILIZER FOR AIRSHIPS

rod 20 by the disk 19, sitting on the steering wheel, and moving as it moves except in the case when the wheel rotates about the vertical axis. The pedal 12 serves to cut out the servomotor, and leaves the aviator free to maintain his equilibrium in the usual way, by operating the lever 17.

The servomotor could be operated either by air or by water. The inventor chose the latter because water, as a highly incompressible fluid, permitted the apparatus to be more quick-acting and compact. The apparatus has not yet been actually constructed.

Fuel and Firing

TEER-KOKSGRUS-UNTERWINDFEUERUNG, BÖNDEL. *Journal für Gasbeleuchtung*, March 9, 1912. 1½ pp., 1 fig. *d.* Description of a *forced-draft furnace* built by J. Schwartzkopff in Lauban, Germany, *for very small coal and gas tar as fuel.* The furnace has a grate consisting of a plate with nozzle-shaped holes bored through it, the holes being set closer at the coking plate and nearer the fire bridge than in the middle of the grate. An undergrate steam-jet blower supplies a powerful draft consisting of a mixture of steam and air, which cools the grate and raises the temperature of the fire better than a blast of air alone. Of extreme importance is a correct arrangement of the steam-jet blower which can be found only by trying; it was found during tests that even very slight changes in the arrangement of the blower affected the efficiency of the apparatus quite seriously. The volumetric efficiency of the draft was found to be proportional to the steam pressure, and when the steam is dry, saturated steam is better than superheated. It was further found by careful experimenting that a battery of small jets requires less steam for the same amount of work done than one big jet. The tar-burning device is arranged so that the tar flows in a thin stream into the combining nozzle, is there caught up by a steam jet and is carried into the furnace in a highly divided state. The amount of steam required is about equal, by weight, to that of the tar consumed.

AXERS NEUE SELBSTSTÄTIGE UND VON HAND BENUTZBARE SCHUR- UND ABSCHLACK VORRICHTUNG, Pradel. *Zeits. für Dampfkessel und Maschinenbetrieb*, March 1, 1912. 1 p., 3 figs. *d.* Description of an *automatic stoker with an automatic grate-cleaning device* invented by E. Axer, of Altona, Germany. The grate bars are laid not longitudinally, as usual, but crosswise, and may be inclined either toward the front or the back. Before reaching the fuel, the air has to pass through two sets of bars, and is therefore better preheated than in the ordinary flat grate. Half of the grate bars is stationary while the other half has a slow motion first one way and then another. This motion is adjustable, and helps better combustion and cooling of the bars, as well as breaking up of clinkers. In fact, the constant motion of the bars leaves no time for the clinkers to form and does not permit them to melt together into large lumps.

Heating

NOTE SUR LE CHAUFFAGE À L'AIDE DE LA VAPEUR DE DÉCHARGE DES MACHINES MONOCYLINDRIQUES OU DE LA VAPEUR PRISE AU RECEIVER DES MACHINES COMPOUND, Lecuir. *Revue de mécanique*, February 29, 1912. 8 pp., 9 figs. *c.*

Comparison of various *systems of steam heating* in connection with steam power plants from an economic point of view.

There may be two cases in accordance with the pressure used in the heating mains, which may be either that of the boiler, or less. In the first case the best plan is to have the steam engines consume as little steam as possible, and to return to the boiler the water of condensation from the heating system by a steam loop: a hot water pump is to be avoided owing to its poor efficiency. If the pressure in the heating mains is less than that in the boiler, one of the following three systems may be used: (*a*) the steam may be taken from the boiler, and the temperature of the steam reduced by a reducer; the work of expansion is, however, lost in

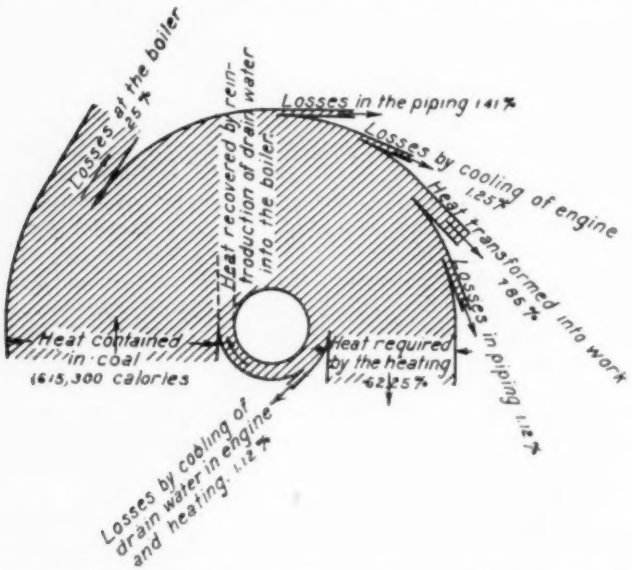

Losses at the boiler 25 %
Heat recovered by reintroduction of drain water into the boiler
Losses in the piping 1.41 %
Losses by cooling of engine 1.25 %
Heat transformed into work 18.5 %
Losses in piping 1.12 %
Heat required by the heating 62.25 %
Heat contained in coal 1613,300 calories
Losses by cooling of drain water in engine and heating 1.12 %

FIG. 4 HEAT BALANCE OF A SINGLE-CYLINDER STEAM ENGINE

this case, and cannot even be considered to be compensated for by the superheating of the steam, because the fall of pressure is not large enough, and the use of superheated steam in heating is but seldom advantageous. (*b*) Installation of a separate boiler having the pressure required for the heating system: this complicates the service in the boiler room, involves large losses in fuel, but is on the whole preferable to the first solution. (*c*) The use of steam which has done work: by such steam the author means steam taken from the exhaust of a single-cylinder engine or the receiver of a compound engine.

Fig. 4 represents the heat balance of a single-cylinder 200-h.p. engine. The heat necessary for heating in the plant is 1,006,500 calories (say 4,000,000 b.t.u.), and if it had to be produced separately, with coal having a heat value of 7000 calories per kilogram (say 12000 b.t.u. per lb.), about 240 lb. of coal per hour would have to be consumed.

Fig. 5 shows the heat balance of a tandem compound engine with heating with live steam. The total economy is considerably less satisfactory than in the case of a single-cylinder engine.

The relation between the type of heating and the machine used may be expressed as follows: If the heating plant requires steam in amount equal or slightly exceeding that furnished by the engines, at a pressure 300 to 600 grams (4.25 to 8.5 lb. per sq. in.) above atmospheric, a single-cylinder engine is advisable, because this pressure corresponds to the final pressure for a diagram showing the best economic utilization of such an engine. If at times a little more steam is required for heating, it may be taken direct from the boiler, and brought down to the required pressure by a reducer. If, however, the quantity of steam required for heating is con-

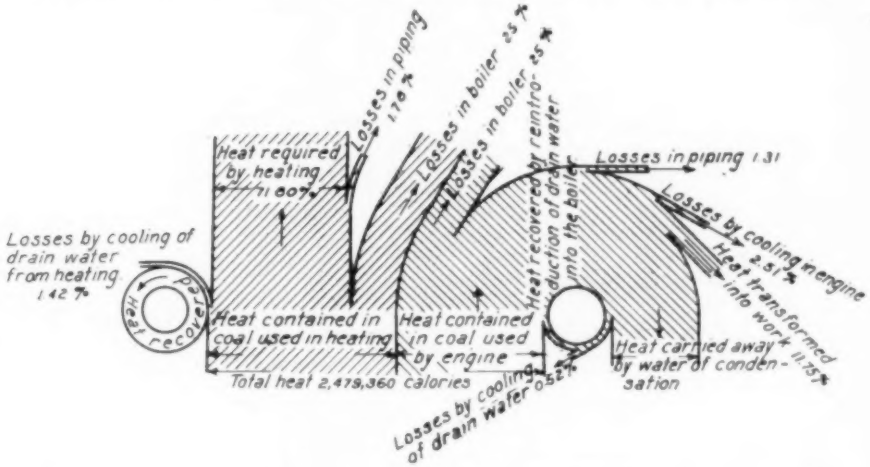

Losses in piping 1.7%
Losses in boiler 25%
Losses in boiler 25%
Heat recovered or reintroduction of drain water into the boiler
Heat required by heating 11.60%
Losses in piping 1.31
Losses by cooling in engine 2.51%
Heat transformed into work 11.75%
Losses by cooling of drain water from heating 1.42%
Heat recovered
Heat contained in coal used in heating
Heat contained in coal used by engine
Heat carried away by water of condensation
Total heat 2,479,360 calories
Losses by cooling of drain water 0.52%

FIG. 5 HEAT BALANCE OF A TANDEM COMPOUND ENGINE WITH HEATING WITH LIVE STEAM

siderably less than that delivered by the engines, or is at a pressure of from 600 grams to 2.50 kg. (8.5 to 35.5 lb. per sq. in.) above atmospheric, a compound engine should be used, with the steam taken at the receiver and constant pressure maintained by varying the admission to the low-pressure cylinder.

This relationship exists also in the case of turbines, the high pressure turbine corresponding to a single-cylinder engine, and the two-stage turbine to a compound engine, the steam in the latter case being taken at the point where its pressure corresponds to that in the heating system.

The author shows by indicator diagrams how steam taken for heating from a compound engine affects the working of the engine. When steam going to the low-pressure cylinder is taken to the heating system, the work that would have been developed in it has to be made up by the high-pressure cylinder, and if much steam is regularly taken in order to insure normal working of the engine, the volume of the low-pressure cylinder

must be decreased. If too much steam were taken, the low-pressure cylinder would receive none, and would give no work; the engine would then work as a single-cylinder engine with back-pressure, driving the low-pressure cylinder at no-load, and its efficiency would be very poor. On the other hand, if too little steam is taken, too much load is thrown on the low-pressure cylinder. There are therefore certain upper and lower limits of admission to the low-pressure cylinder, which must be maintained by some sort of regulation either by hand or automatically, in order that the plant should work at its maximum efficiency. In general the tandem or cross-compound engine lends itself better to the taking of steam from the receiver than a twin-compound engine.

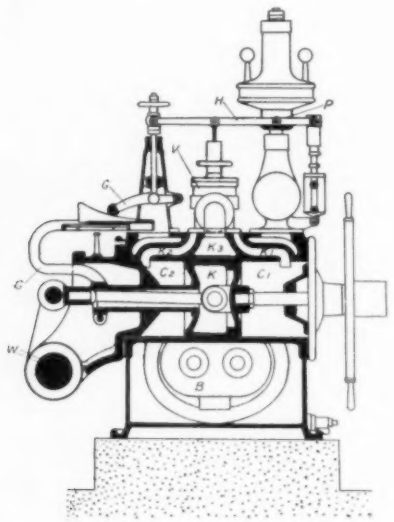


FIG. 6 OIL REGULATOR IN THE BELL WATER TURBINE

Hydraulics

LE TURBINE IDRAULICHE. *L'Industria*, March 17, 1912. 1 p., 3 figs. *d.* Description of the *hydraulic turbines* of the Société anonyme des Ateliers de Construction de Th. Bell et Cie de Kriens (Lucerne, Switzerland). The turbine is of the Francis type. The *regulator* is shown in Fig. 6. The oil under pressure is driven from the reservoir B by a pump, driven by the pulley S, to the valve gear V, which communicates with the cylinder chambers C_1 and C_2 by the ducts K_1 and K_2, and with the reservoir B by the duct K_3. The valve gear is regulated by the spring governor R in such a way that the pressure acting on one side of the piston K varies according to the conditions of work of the turbine, and produces displacements of the piston which are transmitted to the governor spindle. To decrease the stroke of the piston, and to prevent excessive opening and closing of the turbine, the gear rod G is provided to bring the valve into its middle

position, so that when the number of revolutions of the turbine is constant, the pump works as if there were no pressure, and it had only to make the oil circulate, thus making the consumption of power very slight.

Internal-Combustion Engines

PERFECTIONNEMENT DES MOTEURS À COMBUSTION INTERNE PAR LE RECHAUFFAGE PRÉALABLE DE L'AIR, A. Nougier. *Le Génie Civil*, March 2, 1912. 3½ pp., 3 figs. *dt.* Continuation of the article abstracted in *The Journal* for April (p. 615) on the *improvement of internal-combustion engines by preheating of the air* admitted to the cylinders.

The author recommends a preheater constructed on the principle shown in Fig. 7. It is built like a semi-tubular boiler, the hot exhaust gases passing through the flues in one direction, and the air traveling in the opposite direction, the baffle-plates *MN* forcing the air to take the longer path indicated by the arrow line.

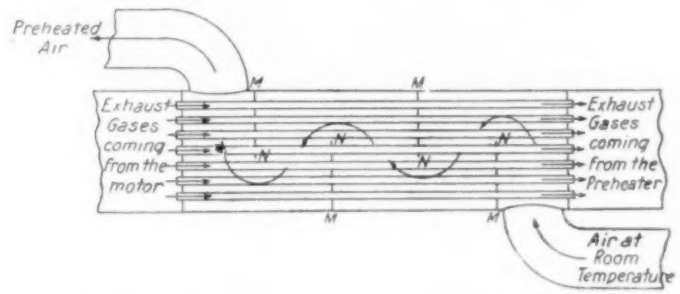


FIG. 7 NOUGIER AIR PREHEATER FOR INTERNAL-COMBUSTION ENGINES

The difficulty in the application of this method lies in the fact that, with a 200-h.p. motor taking in the air at room temperature, the temperature of the exhaust at full load is 466 deg. cent. (870 deg. fahr.), but only 135 deg. cent. (275 deg. fahr.) at no load, and this temperature is too low sufficiently to preheat the air, or, if the preheater were made efficient at this temperature, it would preheat too much at full load, and too high preheating, as was shown in the preceding article, is not economical. By a detailed calculation the author proves that a preheater can be constructed which will preheat the air to 80 deg. cent. (176 deg. fahr.) at no load, and will not overheat the air at full load; this preheater will be only 0.60 m (23 in.) in diameter, and will contain 100 tubes, 21/15 mm (0.055 in.) in diameter and 1.06 m (39.6 in.) long.

The author proceeds to calculate in detail the thermal efficiency of the motor with 35 atmospheres compression with and without preheating, and with 25 atmospheres compression with preheating, and shows that when the compression remains the same, the thermal efficiency of the motor is not affected by preheating, and that the thermal efficiency of the motor with 25 atmospheres compression and preheating is only about 4 per cent less than that of the motor at 35 atmospheres compression without pre-

heating. Since, however, the reduction of compression pressure will give considerable mechanical advantages, very probably exceeding 4 per cent, there appears theoretically to be good ground to believe that preheating of the air will give an over-all increase of efficiency.

Über eine neue Bauart des Kerpely-Gaserzeugers. H. Hermanns. *Dinglers Polytechnisches Journal*, March 9, 1912. 2 pp., 1 fig. *d.* Description and data of tests of a new type of the *Kerpely gas producer* using cheap fuel, and utilizing waste heat for the production of steam. *Cp. The Mechanical Engineer*, March 29, 1912, and *Stahl und Eisen*, December 28, 1911.

Neuere Rohölmotoren. Ch. Pöhlmann. *Dinglers Polytechnisches Journal*, March 30, 1912. 4 pp., 19 figs. *d.* Extensively illustrated description of the Breslau oil motor, with many details on the construction of its cylinder cover and compressor.

Neuere Rohölmotoren, Ch. Pöhlmann. *Dinglers Polytechnisches Journal*, March 16, 1912. 2½ pp., 4 figs. *d.* Description of *high-speed oil motors* built by the Augsburg-Nuremberg Co. and by the Breslauer A. G. für Eisenbahnwagenbau und Maschinenbauanstalt. The latter motor gives 450 effective h.p. at 400 r.p.m.

Motori à combustione interna. *L'Industria*, March 17, 1912. 2 pp., 9 figs. *d.* Description of the Henriod, Cottereau, Broc, and Ballot *slide valve motors* (Cp. *The Journal*, March, 1912, p. 414, and the *Automobile*, August 3. 1911).

Machine Shop

Procédé de fabrication des tubes en U. *La Métallurgie*, March 20, 1912. 2/3 p., 12 figs. *d.* Description of a *process for making U-tubes*, in particular for superheaters, patented in France by W. Schmidt. This process requires no special tools, and is claimed to produce a tube of great homogeneity at the joint. A rather deep groove, 3, as shown in Fig. 8, is cut in the tube, normal to its longitudinal axis, so as to obtain at the place where the U is to be formed two tubes connected by a short strip *c*. The tubes are then bent back about an axis passing through the strip *c*, into the position shown in 4, the distance between the axes of the tubes depending on the width of the groove. Next a piece of flattened tube *e* (9 and 10) is placed on the mandrel *d*, hammered into shape, and the joint above the mandrel welded, so that a sort of cap *f*, open at one end, is obtained. This cap is hammered or pressed over a mandrel into the shape shown in 7 and 8, placed over the tubes *a* and *b*, and welded (1 and 2). The presence of the connecting strip *c* is said to offer the following advantages; no weld is necessary at the narrowest place, i.e. just between the tubes; only two pieces have to be handled in welding; the presence of the strip makes the joint stronger.

Benzol-Schweissung. *Autogene Metallbearbeitung, Beiblatt zu Acetylen*, February 15, 1912. 5 pp., 1 fig. *cd.* Description of *oxy-benzine welding*, and comparison of it with oxy-acetylen welding. The temperature produced by the oxy-benzine flame is about 2800 deg. cent. (5072 deg. fahr.) against 3200 deg. cent. (5792 deg. fahr.) of the oxy-acetylen flame. The

time required for welding with oxy-benzine is longer, from 30 to 100 per cent., in accordance with the thickness of the material. Plates of more than 8 mm. (0.31 in.) cannot be safely welded by oxy-benzine, because benzine is not a chemically pure product, and contains hydrogen which dissolves in iron at the temperature of welding, but separates at cooling, forces its way outwards from the joint, and in doing so makes a path for the escape of carbon; the joint is therefore apt to lose its carbon, and to become hard and brittle. A further disadvantage of the oxy-benzine method

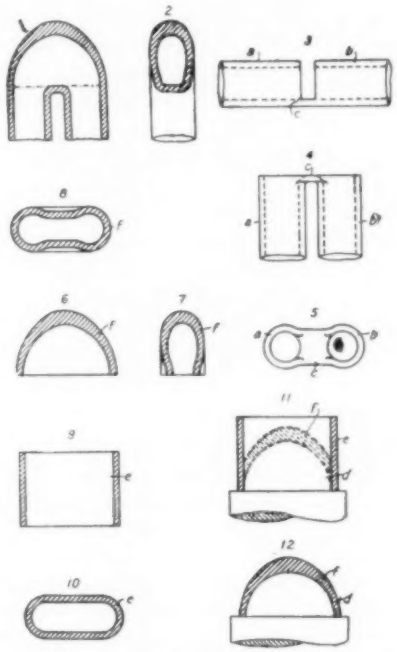


FIG. 8 PROCESS FOR MAKING U-TUBES

of welding consists in the fact that, as has been shown experimentally, the quality of work is substantially affected by the temperature and conditions of work in the place where the welding is carried on; at low temperatures (below 4.5 deg. cent., or 40.1 deg. fahr.) the benzine freezes, and must be either heated or mixed with alcohol, the latter unfavorably affecting the strength of the joint produced; drafts of air are liable to change the proportion of benzine and air mixture in the carburetor spiral, and that in its turn affects the quality of the work done. Benzine fumes are to a certain extent dangerous to the health of the workmen, although the products of its combustion are harmless. The use of oxy-benzine welding for regular shop or factory work cannot therefore be recommended, but the compactness of the apparatus and practical safety from explosions make it very convenient for road and erecting work. Fig. 9 shows the comparative cost

of oxy-benzine and oxy-acetylen welding per meter (or 39.37 in.) of welded seam in pfennigs, the plates welded being in millimeters (1 mm equal to 0.039 in.), and the basis of comparison being: acetylen at 1 M. per cu. m., or, say, 0.71 cents per cu. ft.; benzine, 90 per cent pure, 24 pfennigs per kilogram, or, say, 2.7 cents per lb.; oxygen and labor in both cases, the first 1 M. per cu. m., or 0.71 cents per cu. ft., and the second 60 pfennigs, or 15 cents, per hour.

PROCÉDÉ ET APPAREILS POUR LA FABRICATION DES PLAQUES MÉTALLIQUES. *La Métallurgie*, March 13, 1912. 1 p., 3 figs. *d.* Description of a new *process for galvanizing metallic objects*, invented by Kar Miehl of Gutersloh, Germany. The objects to be galvanized are placed on a special receptacle provided with holes like a screen, and are dipped into a bath of galvanizing metal, and kept there until they attain a temperature very

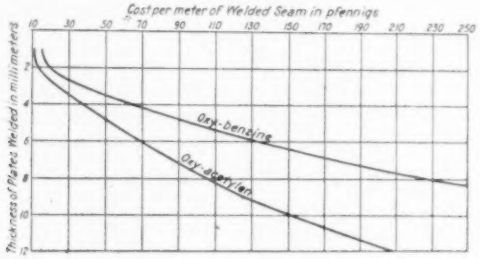


FIG. 9 COMPARATIVE COST OF OXY-BENZINE AND OXY-ACETYLEN WELDING

near that of the bath. The receptacle with the objects on it is then quickly transferred to a drum which is set into rapid rotation. Owing to the action of centrifugal force, in a fraction of a minute all the excessive fluid material is projected outside of the receptacle, and the objects remain covered by a uniform and closely adhering layer of metal. This process permits of galvanizing very small objects, e.g. needles, as well as such objects as screws, which could not be galvanized by simple immersion in hot metal because the metal would fill the screw thread. The article also describes the simple machinery used in this process.

Mechanics

NOTE SUR LE CALCUL DU TRAVAIL DU MÉTAL DANS LES CABLES MÉTALLIQUES, Edgar Baticle. *Annales des Ponts et Chaussées*, January-February, 1912. 20 pp., 9 figs. *mtA.* Mathematical investigation of the causes producing rapid *deterioration of wire ropes*. The author gives formulae for determining the stresses in the rope when its radius of curvature is known, and shows how to determine, in load ropes, the sum of the components of bending moments and moments of torsion. He concludes that it is possible by the usual methods of determining the resistance of materials to find the total stress in the most stressed fibers of a rope, whether wound about a drum of known radius, or subjected to a concentrated load which gives it a certain curvature of unknown radius. This total stress is composed of the stress of tension, bending and torsion stresses, the last being smaller in a

well made rope. The bending stress in the wires composing the rope is due to the following two causes: (*a*) in the manufacture of the rope the wires are often subjected to considerable stresses resulting in their losing some of their power of resistance; (*b*) the influence of the bending of the rope under a live load, or while coiled on a drum. The stress due to the last cause is variable. If, as we believe generally happens, the total stress exceeds the limit of elasticity, the variable and repeated stress due to bending must rapidly cause plastic deformation in the metal. The stress produced in manufacturing the rope varies inversely as the radius of "elementary wire." The same is true for the bending stress in a rope coiled on a drum of a definite diameter, but in load ropes the stress is independent of the diameter of the elementary wire.

From the method applied by the author it follows that, as regards deformation, a rope may be considered as a member under flexure, provided an imaginary moment of inertia is attributed to it, of which the author gives the value as a function of the total section of the rope, radius of the elementary wire, and angles of coiling of the wires and strands. The flexibility of a rope is measured by the reciprocal of this imaginary moment of inertia, and is inversely proportional to the radius of the elementary wire and radius of the rope, but varies with the method of making the rope, and is generally greater in a rope of several strands than in a rope made of a single strand.

THEORETISCHE BERECHNUNG EINER SCHLEUDERPUMPE AUF GRUND VON VERSUCHEN, H. C. A. Ludwig. *Die Turbine*, Nos. 1 to 12, 1911 to 1912. *t*. The author shows from a test of a *centrifugal pump* how the data obtained experimentally may be brought into agreement with theoretical values. He draws the usual *O-H* curves from an empirical formula, and proves that these curves agree with the values for impacts of water obtained in accordance with the usual theory. He further compares his *O-H* curve with those of other investigators, and shows their material agreement.

ÉTUDE GRAPHIQUE DE LA RÉSISTANCE DES PIÈCES À FORTE COURBURE RELATIVE, F. Legein. *Bulletin technique de l'Association des ingénieurs sortis de l'École polytecnique de Bruxelles*, February, 1912. 17 pp., 6 figs. *mA*. Prof. W. Ritter (*Der elastische Bogen*) has graphically studied the *resistance of arches* of small curvature. The author applies his method to the case of members *of large curvature*, and, by way of example, shows how to apply it to the investigation of a ring, of circular section, and radius of curvature equal to *2r*, acted upon by an axially directed force.

VERSUCHE ÜBER DIE STRÖMUNGSVORGÄNGE IN ERWEITERTEN UND VERENGTEN KANÄLEN, H. Hochschild. *Mitteilungen über Forschungsarbeiten auf dem Gebiete des Ingenieurwesens*, Vol. 114, 1912. 53 pp., 70 figs. and 27 tables. *c*. Account of an investigation of the *conditions under which flow takes place in expanded and contracted passages*, such as are met with in turbines and turbo-blowers. The theory of Prof. L. Prandl was laid as a basis of investigation, and the author quotes from his manuscript as follows: if two particles of a fluid glide over one another in such a manner that in a distance *dy* there is a difference of velocity *dw*, there is produced in the surfaces of gliding contact a shearing stress $t = k \frac{dw}{dy}$ The con-

stant k is called the coefficient of internal friction, or viscosity. The author's experiments have throughout corroborated this theory. Measurements of hydraulic pressure have shown that in contracted passages the flow occurs with nearly no losses (potential flow), and as the passage expands, the conditions of flow become less favorable. Further investigations by means of a special little tube (arranged somewhat like a pitot tube) have shown that the losses, in the form of eddies formed by friction, arise mainly at the walls of the passage, and gradually permeate the whole stream ("*Turbulenz*," whirls), and that in direct proportion to the expansion of the passage and rate of decrease of velocity and increase of pressure.

VERSUCHE MIT RIEMEN BESONDERER ART, Kammerer. *Zeits. des Vereines deutscher Ingenieure*, February 10, 1912. 7 pp., 15 figs. *cA*. Account of tests on the *limits of effective stress in belts* of various materials. Such tests require a large expenditure of time because overloading of a belt can be discovered only from the appearance of a permanent elongation, and an identical elongation is also found in a new belt which has not yet reached its state of resistance. Since, however, several hours are required for a belt to reach that state, there is always a danger in short tests with new belts of taking phenomena of the state of resistance for those of overload.

Tests of Link Belts. The tests were made on two link belts of equal width, but unequal thickness, made up of links pressed out of specially prepared fiber, and provided with leather rims. The tests have shown that at a speed of 15 m (say 50 ft.) per sec., the total stress on the tight side of the belt must not exceed 30 kg per cm (166 ft. per in.) of width of belt, but at higher speeds may go as high as 35 kg per cm (193 lb. per in.) of width of belt. Link belts have been introduced because they combined the strength of double with the flexibility of single belts, and gave strong belts which could be used on pulleys of small diameter. These tests have shown that for speeds up to 20 m (65 ft.) per sec. link belts are far superior not only to single, but even to double, leather belts, but at higher speeds the centrifugal tension in the heavy link belt decreases the allowable useful load very materially, and at speeds over 30 m (98 ft.) per sec. link belts cannot be economically used at all.

Tests of Leather Belts. For speeds up to 15 m (50 ft.) per sec. the total stress in the tight side of the belt did not exceed 23 kg per cm (say 126 lb. per in.) of the width of the belt, while for speed of 30 m (98 ft.) per sec. the total useful tension rose to 26 kg per cm (143 lb. per in) of width of belt.

Americans often assert that belts which are turned with the *hair side to the pulley* last longer than when turned with the flesh side to the pulley, as is usual in Germany. Tests made by the author have shown that in belts turned with the hair side to the pulley: (*a*) the tension ratio, or ratio between the tension on the tight side and that on the loose side hardly exceeds 2, while in belts running with the flesh side to the pulley it can exceed 3; (*b*) the highest allowable tension is more than 3 kg per cm (12.6 lb. per in.) less than in the case of belts with the flesh side toward the pulley; the speed cannot be made more than 30 m (98 ft.)

per sec., against 50 m (164 ft.) with belts with the flesh side toward the pulley. This shows that it is in every respect disadvantageous to let the belt run with the hair side toward the pulley. A belt running in that way probably wears out sooner because, as was pointed out by C. O. Gehrkens, such curvature reverses the way in which the hide grew on the animal.

Tests of High-Speed Belting. A double cemented leather belt, 45 mm (1.8 in.) wide, stood very well a total tension on the tight side of 67.1 kg per cm (370 lb. per in.), at 60 m (196 ft.) per sec., and broke only when the tension rose to 95.9 kg per cm (say 530 lb. per in.). The allowable useful stress k_n can be deduced from the formula

$$k_n = (k_T - k_f) \times \%$$

where k_T is the total stress on the tight side, and k_f the centrifugal tension.

Tests on Belt Fastenings. In the tests with woven belts it was found that, as the speed of the belt transmission increased, the allowable useful stress diminished owing to the influence of the belt fastening, which at speeds from 25 to 30 m (81 to 98 ft.) per sec. caused such vibrations and uncertainty of run that it proved to be impossible to exceed the speed of 30 m (98 ft.) per sec. The influence of the fastening is due to the following causes: (*a*) through the center of gravity of the fastening describing on the pulley a semicircle, an additional centrifugal tension is developed equal to

$$k'_f = \frac{G}{bl} \frac{v^2}{g}$$

this additional centrifugal tension is thus a function of $\frac{G}{bl}$, or weight of unit area of the fastening, and that indicates the advisability of arranging the fastening in such a way that its weight be distributed over as large an area of the belt as possible; (*b*) the rotation of the fastening in coming on and off the pulley produces in it a moment of torsion which in its turn produces in the belt an additional centrifugal tension. This tension becomes negligible if the fastening is made of flexible steel bands, softer steel bands lasting longer than hard ones.

The article also contains data on tests of woven belts, and discussions on the theory of belting which could not be reported owing to lack of space.

Refrigeration

DIE VERBESSERUNG DER KOHLENSÄURE-KÄLTEMASCHINE DURCH EINFÜHRUNG EINES EXPANSIONZYLINDERS, R. Plank. *Zeits. für die gesamte Kälte-Industrie*, March 1912. 3 pp. *t.* Discussion of the improved *carbon dioxide refrigerating machine* with a separate expansion cylinder proposed by L. Horst (see *The Journal*, February, 1912, p. 294). Plank finds that the machine is based on correct thermodynamic principles, and represents a real improvement on the present CO_2 refrigerating machines, but not as large an improvement as Horst claimed.

Steam Engineering

ZUR STREITFRAGE DER GLEICHSTROM-DAMPFMASCHINE, E. Tuckermann. *Dinglers polytechnisches Journal*, March 9, 1912. 4 pp., *ct.* Comparison

of the *straight-flow steam engine*, as designed by Stumpf, with other similar types, and discussion of its mechanical characteristics. The peculiar property of the straight-flow steam engine is that, as a single-cylinder engine, it works with saturated steam just as economically as with superheated, and in heat economy reaches a degree as high as, or higher than, large multi-cylinder engines with superheat. That does not mean that superheating the steam is of no value whatever. On the contrary, all along the way up to the admission valve the economy of superheat even in straight-flow steam engines cannot be denied, but, once inside the cylinder, dry saturated steam gives as good results economically as superheated, because it does not require as expensive a lubrication system as an engine for superheated steam. The working cycle of the straight-flow steam engine approaches more the Carnot cycle than does that of an engine working with superheated steam, the admission line in the first case being more of the nature of an isotherm, and for the latter of an isobar, of the gas. For equal heat losses during the admission periods in both engines, the pressure in the superheat engine falls more rapidly than in the saturated steam engine. The value of superheating lies in the fact that it compensates for losses causing the performance of the engine to depart from that of the ideal engine, and if these deviations are overcome in some other way, superheating becomes superfluous.

There is a substantial difference between a straight-flow and alternating-flow steam engine in the working of the exhaust. Suppose that in both cases the steam at the end of expansion is wet. In a straight-flow engine it becomes immediately dry, because it suddenly expands without doing any work, and thus preserves its heat, while the pressure goes down. The piston head is therefore dry at the beginning of compression, and the heat flowing from the walls and especially from the hot cylinder cover, highly superheats a large amount of steam. In an alternating-flow steam engine the exhaust lead is not as rapid, and the piston drives part of the wet steam toward the exhaust, and owing to the high velocity of the steam flow, cools both the admission and exhaust openings: the exhaust may be said to come out cold. The piston, which is wet, remains all the time in contact with the cold exhaust steam, and more heat is required for evaporation than in the case of a straight-flow engine. The average difference of temperature between the walls and exhaust steam, and consequently the heat given off to the exhaust steam, is greater in the case of an alternating-flow engine, but all that heat goes into exhaust, and is not utilized as in the straight-flow engine.

The valve gear of the straight-flow steam engine is remarkably simple, owing to the absence of the exhaust gear. The exhaust area is made as large as practicable, an ideal equilization of pressures between the cylinder and condenser being simply a result of the constructive peculiarities of the engine.

The use of a long piston is generally pointed out as one of the disadvantages of the straight-flow engine. The author argues that long pistons have been used quite successfully in gas-engine construction, and that it ought to be all the more easier to use them in steam engines, especially such as do not make use of high superheats.

ABWÄRMEVERWERTUNG AN SIEMENS-MARTIN-OFEN ZUR DAMPFERZEUGUNG, A. Pfoser. *Stahl und Eisen*, March 7, 1912. 1 p., *p.* Discussion of the *utilization of heat of gases discharged from a Siemens-Martin furnace.* The temperature of the gases is 600 to 700 deg. cent. (1112 to 1292 deg. fahr.), and the heat losses are therefore very large. The plant investigated by E. Mayer (Die Wärmetechnik des Siemens-Martin-Ofens) has, in round figures, the following heat balance:

	Per Cent
Total useful work	27.0
Losses by radiation of the furnace and regenerators	29.0
Losses by exhaust	31.0
Combustible in the ash-pan of the gas producer	3.0
Losses by radiation of the gas producer	10.0
Heat produced by the combustion of coal	100.0

The author shows that by using the discharged gases to produce steam at 10 atmospheres pressure above atmospheric, which can be done by reducing the temperature of the exhaust gases to 300 deg. cent. (572 deg. fahr.), the efficiency of the furnace can be improved 16.7 per cent. The important problem is, however, to have the steam-producing plant designed in such a way as not to interfere with the work of the Siemens-Martin furnace, which after all is the most important part. To do this, the smokestack must be large enough to give a sufficient partial vacuum to overcome the increased friction due to the lengthening of the path of the gases. When artificial draft is used, an economizer may be installed behind the boiler, the temperature of the escaping gases reduced 150 to 200 deg. cent. (302 to 392 deg. fahr.), and about 20 per cent more steam produced. The author claims to have constructed successfully working installations of this kind in connection with zinc and glass melting furnaces.

LES AVARIES DES PLAQUES TUBULAIRES DANS LES CHAUDIÈRES DE LOCOMOTIVES, H. Lavialle d'Anglards. *Le Génie Civil*, March 9, 16, 1912. 4 pp., 10 figs. *p.* Investigation of causes of *breakdowns in tube plates of locomotive boilers*, and description of methods of repair, both on the road and in the shops. If the plate broke between two or three flues, and there is no time to send the locomotive to the shops, the flues are removed, and the hole covered by a buffer plate; the plate is generally placed on the inside of the boiler, with the screws on the furnace side. If more than two or three flues are affected, putting them out of business might cause serious inconvenience, and therefore the solid buffer plate is not used then; instead, the diameter of the end of the flue is reduced, a ring (steel, bronze, or red copper) screwed into the slot so as to cover the broken part of the plate, and the flue hammered over the ring.

The tube plates break down particularly often on locomotives working on lines of uneven profile. When going up a grade, the exhaust is strong, the fire active, the flue tubes get hot and tend to expand, which they can do only toward the inside of the boiler owing to the rigidity of the tube plate; as a result, the plate is under strain. When the train goes down grade, the temperature of the tubes rapidly falls, especially if the injector is set into action, they suddenly contract, and exert a powerful strain on the plate in an opposite direction. These rapid changes of

stresses acting on hot metal produce in it fatigues, and finally lead to breakdowns.

To preserve the plates the author recommends: preheating of feed water (tried on two series of locomotives, but too recently to have any results recorded); superheating (in trials on a large French system it was found that the locomotives without superheat had their tube plates all cracked after a run of 150,000 kilometres (say 93,000 miles), while the plates in locomotives with superheat were intact); choice of suitable materials (on the Saxony railroads good results were obtained with the upper part of the tube plate made of steel, while the lower was of copper, to better resist the action of the fire).

LA TURBINA À VAPORE "TOSSI" MARINA, Fr. Modugno. *Rivista Marittima*, February 1912. Description of the *Tossi marine turbine*. (Cp. *The Engineer*, London, July 22, 1910).

DIE ANFANGSTEMPERATUR ALS EINE WERTVOLLE KONTROLLE BEI VERDAMPFUNGSVERSUCHEN. O. Binder. *Chemiker-Zeitung*, March 14, 1912. 2 p. 1. Discussion of *initial temperature as a valuable method of control in evaporation tests*. The total heat evolved during combustion is at the moment of combustion contained in the gases of combustion. Therefore, if the initial or combustion temperature, the volume of gases of combustion, and their specific heat is known, it is easy to calculate the heat of combustion, and this must be equal on one hand to the sum of the heat in the steam, the heat in the exhaust gases, and heat losses by conduction and radiation, and on the other hand to the theoretical heating value of the fuel, minus the latent heat in the residues of combustion and unconsumed gases and soot. The losses by conduction and radiation may be determined by heating the boiler plant up to a certain steam pressure, removing the fire, and closing the fire doors and dampers air tight. The losses by conduction and radiation can then be determined from the fall in steam pressure after some little time.

The author has shown elsewhere (*Mitteil. über Gegenstande des Artillerie-und Geniewesens* 1911, vol. 2) that the initial temperature may be found from the following formula:

$$T = \frac{\text{Cal.} - w}{CO_2(0.4886 + 0.00024\,T) + N(0.308 + 0.00007\,T) + H_2O(0.4692 + 0.00015T)}$$

where *Cal.* is the theoretical heating value of the fuel, w the undeveloped heat; CO_2 volume of carbon dioxide evolved from 1 kg. of fuel; N volume of nitrogen, oxygen, and other gases, having the same specific heat, evolved from 1 kg. of fuel; H_2O volume of steam corresponding to 1 kg. of fuel.

The heat required to raise the temperature of the gases of combustion by 1 deg. cent. is a function of their volume and specific heat. The general formula for initial temperature is $\frac{WE}{v.\sigma}$, where WE is the total heat developed in WE (1 WE is equal to 3.97 b.t.u.), v volume of gas, and σ its specific heat. The total heat developed can be found from the theoretical heating value of the fuel, by subtracting from it the latent heat in the residues of combustion, soot, and unburned gases. The volume of the gas can be computed from the contents of carbon dioxide in the gas, and the analysis of the fuel, in the usual way. Thus,

from the data of a test, in 1 cu. m. of gases of combustion there is $0.5364 \times 0.097 = 0.05203$ kg. of carbon (1 cu. m. of carbon dioxide contains 0.5364 kg. of carbon). But 1 kg. of coal contains 0.6337 kg. of carbon, and therefore from 1 kg. of coal $\frac{0.6337}{0.05203}$, or 12 cu. m. of gases of combustion are evolved. The total heat evolved can therefore be expressed by the following quadratic equation:

$$T \times CO_2 \times 0.4886 + 0.00024T^2 + T \times N \times 0.308 + 0.00007T^2 \times N + H_2O \times T \times 0.4692 + H_2O \times T^2 \times 0.00015 = WE$$

or, more generally:

$$T.a + T^2.b = WE; \qquad T^2 + \frac{c}{b}T - \frac{WE}{b} = 0$$

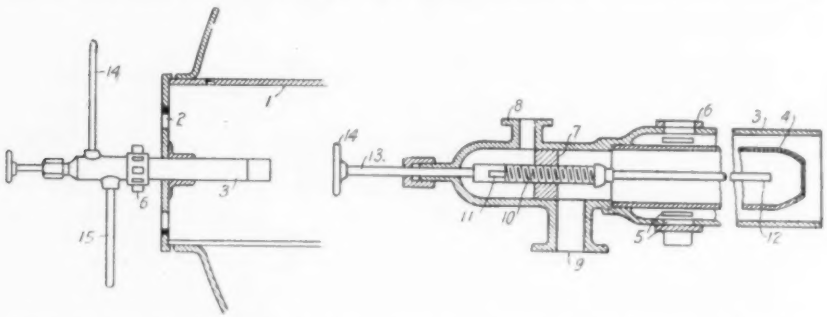
The author proceeds to apply his formula by calculating the data from four tests, and shows that the values obtained by his method are very close to the theoretical heating values.

Verfahren zur Erhöhung des Nutzeffektes von Feuerungsanlagen. *Zeits. des internationalen Vereines der Bohringenieure und Bohrtechniker*, March 15, 1912. 2 pp., 8 figs. *d.* Description of the process invented by Max von Eberhardt for *improving the efficiency of furnaces.* The inventor started from the well-known fact that when combustion is proceeding with air as the supply of oxygen, nearly four-fifths of that air consists of nitrogen which does not help combustion, but takes up a large amount of heat which otherwise might do some useful work. The *inventor therefore brings oxygen to the fuel in the form of superheated steam,* i.e. in connection with hydrogen, the advantage being that, after the steam has dissociated into its component parts, the hydrogen unites with some of the carbon and forms hydrocarbons which materially increase the thermal result of combustion. In fact, the author claims that so high a temperature is obtained that air, and consequently nitrogen, have to be introduced into the furnace as a cooling medium, to make the process practicable for ordinary boiler-plant work. The article describes how this method can be adapted to flue boilers, and oil and gas-fired furnaces. Only the first is abstracted in this article.

The apparatus for use with flue boilers (Figs. 10 and 11) is set as shown in Fig. 10, 1 being the flue, and 2 the flue wall. It consists (Fig. 11) of a casing 3, open at one end, containing a tube 4 somewhat contracted in its forward end. All around the casing there are slots 5 which can be more or less opened or closed by the slots on the ring 6 set on the outside of the casing. The inside of the casing 1 is divided by the bearing 7 into two chambers respectively connected with the junction-pieces 8 and 9. In the bearing 7 is located a hollow threaded spindle provided with slots 11, and carrying at one end a pipe 12 open in front, and at the other end a solid spindle 13; this spindle passes through a stuffing box in the wall of the casing, and is provided with the hand-wheel 14. (Fig. 10). The steam is conducted to opening 8 through the pipe 14 (Fig. 11), the gas to opening 9 through the pipe 15, while the admission of air is regulated by the position of the ring 6. The temperature of the flame is regulated to a certain extent by the mutual position of the ends of the pipes 12 and 4, since the rate of diffusion of the steam, and therefore the amount of hydrogen

brought into the flame is inversely proportional to the distance between the ends of the two pipes. The author does not explain how he succeeds in effecting the dissociation of steam into its component parts, but states that tests made with gas and oil furnaces "have shown the correctness of the theoretical calculations."

DIE 41. DELEGIERTEN- UND INGENIEUR-VERSAMMLUNG DES INTERNATIONALEN VERBANDES DER DAMPFKESSEL-UEBERWACHUNGS-VEREINE. *Glückauf*, March 23, 1912. 8 pp. *g*. Report of the 41st meeting of the delegates of the International Association of Boiler Inspection Societies. Among other things was discussed the best material for *stop-valves for superheated steam*. The expectations entertained as late as 1906 that the cast-iron valve would be very soon forced out of business did not materialize. It is cheaper, and that is an important consideration at a time when boiler



FIGS. 10 AND 11 EBERHARDT APPARATUS FOR FIRING WITH STEAM AS SOURCE OF OXYGEN

manufacturers justly complain of reduced profits. In small valves, i.e. valves with walls not thicker than 10 mm (0.37 in.), cast iron proved to be even preferable to steel, because thin steel castings, unless made of nickel steel or other suitable alloys are not reliable. Valves of section of opening of 150 mm (5.7 in.) or over, ought to be made exclusively of steel. The report contains a detailed discussion of the heating of stop-valves, and standard rules for their construction.

DAS INTERFEROMETER ZUR UNTERSUCHUNG DER DAMPFKESSELRAUCHGASE, II. L. *Braunkohle*, March 22, 1912. 7 pp., 9 figs. *d*. Description of new apparatus of Carl Zeiss in Jena, Germany, for *optical analysis of gases of combustion* by means of an interferometer. The analysis is based on the fact that the refractive power of the gases of combustion depends on the contents of CO_2, and therefore a gas refractometer could be constructed permitting the reading of the analysis of a gas from a scale by simple inspection. The article describes several types of apparatus: a simple, portable, apparatus for boiler-room use, exact to within 0.1 to 0.2 of 1 per cent, and a more complicated one for laboratory use, exact to within 0.01 of 1 per cent. From the description the handling of the apparatus appears to be comparatively simple.

Strength of Materials and Materials of Construction

EISENPORTLANDZEMENT IM VERGLEICH ZU PORTLANDZEMENT. H. Passow. *Stahl und Eisen*, March 21, 1912. 3½ pp. *c*. In 1909 the Prussian Ministry of Public Works admitted *slag cement for use in public works* equally with portland cement. This was done after exhaustive tests at the Royal Laboratory at Gross-Lichterfelde West which showed the equality of both kinds of cement as far as strength and other qualities were concerned, the only point on which slag cement was in some instances found to be inferior to portland cement being in its resistance to the action of the atmosphere, but even that only to a very small degree. The commission in charge of these tests made arrangements for 5 and 10-years' tests, the first of which have just been finished. The article gives extensive abstracts from the reports of the commission in charge of the tests (the complete report will not be published for some time), the final conclusion being that, as materials of construction, slag cement and portland cement are of equal quality.

VERSUCHE MIT UMSCHNÜRTEM GUSSEISEN. SYSTEM DR. V. EMPERGER. *Beton und Eisen*, March 11, 1912. 3 pp., 20 figs. *dc*. Continuation of the article on "*strapped*" *cast iron*, i.e. cast iron surrounded by a concrete jacket, in its turn reinforced by an outer helically wound steel wire. The axial distance between the turns of the wire must be at least equal to the thickness of the concrete core, and higher compression strength can be obtained when the wire is wound still closer.

VERSUCHE ÜBER DIE VERDREHUNG VON STÄBEN MIT RECHTECKIGEM QUERSCHNITT UND ZUR ERMITTELUNG DER LÄNGS- UND QUERDEHNUNG AUF ZUG BEANSPRUCHTER STÄBE. O. Bretschneider. *Zeits. des Vereines deutscher Ingenieure*, February 17, 1912. 6 pp., 10 figs. *e*. Account of experiments on torsion in bars with rectangular cross-section, and their elongation and reduction of area under tension. It was found that in low carbon steel bars with the ratio of sides from $h:b = 1:1$ to $h:b = 10:1$, the angle of torsion within the limit of elasticity, can be found from the formula

$$\theta = \psi^1 M_d \beta \frac{b^2+h^2}{b^3h^3}$$

where for ψ^1 must be substituted the values calculated for different ratios of sides by Saint-Venant, and reproduced by the author in one of the tables, and M_d is the moment of torsion, β the coefficient of shear, b the shorter, and h the longer side of the rectangular cross-section of the bar. The largest difference between the values obtained from tests and those calculated from this formula does not exceed 1.5 per cent. For bars with ratio of sides of cross-section from $h:b = 1:1$ to $h:b = 6:1$ according to tests, for ψ^1 may be substituted

$$\psi^1 = 3.645 - 0.06\frac{h}{b}.$$

The article also contains a discussion of the distribution of stresses in the section of the bar, with 4.5 substituted for the values of ϕ given by Saint-Venant in the equation for maximum shearing stress, and a discussion of relation of elongation to reduction of area in bars under tension.

PRÜFUNGSMASCHINE VON 3000 T DRUCKKRAFT FÜR EISENKONSTRUKTIONSTEILE, Ad. Seydel. *Stahl und Eisen*, March 7, 1912. 3 pp., 1 fig. *d.* Description of a hydraulic *machine for testing the strength of materials* used in bridge construction, capable of exerting a pressure of 3000 tons on bars up to 15 m (say 49 ft.) long, and a tension of 1500 tons on bars up to 13 m (say 42 ft.) long. The machine was constructed by the German Association of Bridge and Railroad Supply Manufacturers, with financial help from the Prussian Government and other associations. The experimental work and distribution of data are in charge of a special board consisting of professors of technical schools and practical men.

ÜBER DIE MESSUNG GROSSER KRÄFTE IM MATERIALPRÜFUNGSWESEN. *Dinglers Polytechnisches Journal*, March 9, 1912. 1 p. *t.* Account of a paper read, December 21, 1911, by A. Martens before the Prussian Royal Academy of Sciences on the measurement of large forces in testing of materials and calibration of testing machinery. One of the most sensitive methods appears to be that of G. Wazau: a cylindrical test piece, subjected to tension or compression, is placed in a vessel filled with a liquid, and the change of volume due to the elongation of the test piece is compensated for by immersion of a body actuated by a micrometer screw in such a way that the column of mercury in a capillary tube connected with the main vessel remains always at the same level. This apparatus allows the measurement of the power applied within 2 kg (4.4 lb.) when the load applied is 10,000 kg (22,000 lb.). It is, however, necessary in tests with this apparatus to look out very carefully for possible errors due to variations of temperature which materially affect the readings.

ÜBERBLICK ÜBER DIE GEBRAUCHLICHSTEN FESTIGKEITSPROBIERMASCHINEN, W. Muller. *Dinglers Polytechnisches Journal*, March 2, 1912. *d.* A discussion and description of the usual *testing machines*, such as concrete testing presses, tension testing machines of Werder, Martens, Amsler, and Pohlmeyer, and special machines for hardness and torsion tests.

LE TENSION-MÈTRE. Captain Largier. *Mémoires de la Société des Ingénieurs civils de France*. December 1911. 14 pp., 8 figs. *d.* Description of an *apparatus for measuring tension in wires in active service*. Tightening of wires by turn-buckles necessitated the use of some apparatus that would show when the tension in the wire was approaching the limit of safety, especially since for wires of small diameter a very few number of turns of the tightening screw may mean an increase of thousands of pounds in tension per square inch of cross-section. The very simple apparatus of Captain Largier is based on the formula for the transversal vibrations of strings

$$NL = K_1 \sqrt{t}$$

where N is the number of vibrations per second, L the length of the vibrating string, K_1 a numerical coefficient, and t tension per unit area of cross-section of the string. The apparatus consists of a double rule carrying a resonator and two brackets, one fixed, and the other movable. To find the tension in a wire, e.g. in a stay-wire of an aeroplane, the apparatus is placed on the wire by special grips pressing it against the brackets, and the wire is struck by a piece of wood or pencil so as to make it vibrate; the

movable bracket is then adjusted so that the wire should give a certain definite tone, e.g. *la* (*A* of the American scale). A tuning fork may be used to identify the tone if necessary. Since N is known for a given tone, the formula reduces to

$$L = K_2 \sqrt{t}$$

where K_2 is a numerical coefficient. L and t stand therefore in a parabolic relation to each other, and the rule may be graduated in such a way that the tension in the wire may be read directly from the position of the movable bracket. A correction has to be made, however, in the case of wires of comparatively large diameter. The article contains a curve showing how the increase in the diameter of the wire affects the reading of the instrument.

Thermodynamics

Über die Temperaturänderung von Luft und Sauerstoff beim Strömen durch eine Drosselstelle bei 10° C und Drucken bis 150 Atmosphären. Emil Vogel. *Zeits für die gesamte Kälte-Industrie*, February 1912. 4 pp. *et.* Account of an experimental investigation of the *cooling effect produced in air and oxygen by flow through a throttling orifice*, at 10 deg. cent. (50 deg. fahr.) and pressures up to 150 atmospheres. The experiments were made with a constant difference between the high and low-pressure side of the opening of 6 kg/qcm (85.32 lb. per sq. in.), and with as good protection against heat losses as possible.

It was found, both for air and oxygen, that the cooling effect diminishes as the absolute pressure rises, and that there is for both gases a pressure limit beyond which the expansion to a lower pressure produces not cooling, but rise in temperature. By applying the law of $\frac{1}{T^2}$ for the reduction of temperature discovered by Thomson and Joule, the author obtains the following formulae:
for air

$$dt = (0.268 - 0.00086\ p)\left(\frac{273}{T}\right)^2 dp$$

for oxygen

$$dt = (0.313 - 0.00085\ p)\left(\frac{273}{T}\right)^2 dp$$

where dt is the cooling effect, dp the difference in pressure between the high and low-pressure sides of the opening, p the absolute pressure, taken as an arithmetical mean between the pressures on both sides of the opening, T the absolute temperature taken as an arithmetical mean between the temperatures on both sides of the opening.

It was found that, especially with low pressures, use of poor insulators between the high and low-pressure sides of the opening was responsible for quite misleading results. When hard rubber insulation was used, the relation between increase of temperature and decrease of cooling effect was found to be linear, while with porcelain and especially copper insulation not only the values of cooling effect appeared to be generally less.

but the cooling effect increased with pressure, reached a maximum at about 20 atmospheres, and then began to decrease. This can be explained by the discharge of heat through the walls of the throttling apparatus which, when the amount of gas flowing through the orifice is small, reaches the inner layers of the stream of gas, and consequently the thermometer, and thus apparently diminishes the cooling effect for low pressure. The author shows further that his formula agrees with the results obtained by Thomson and Joule, and Linde.

The inversion point, i.e. pressure beyond which the expansion of the gas through a throttling orifice produces rise of temperature instead of fall, at 0 deg. cent., is as follows:

	From van der Waals' equation.	From author's formula.
	In atmospheres.	
Air	320	312
Oxygen	362	370

This can be also expressed in the following way. The author's experimental formula may be expressed in the following general form:

$$dT = \frac{a - bp}{T^2}\, dp$$

If the gas is expanded from a higher pressure p_1 to a lower pressure p, the temperatures of the gas at the respective pressures being T_1 and T, the cooling will be found by an integration to be

$$\delta = T_1 - T = T_1 \sqrt{T^2 - 3(p_1 - p)a + \frac{3}{2}(p_1^2 - p^2)b}$$

and therefore the following conditions may be deduced:
cooling of expanding gas if

$$p_1 + p < \frac{2a}{b}$$

no change in temperature if

$$p_1 + p = \frac{2a}{b}$$

rise of temperature if

$$p_1 + p_1 > \frac{2a}{b}$$

These phenomena cannot be explained by external work, and must therefore be ascribed to the action of intermolecular forces. The technical importance of the investigation lies in its relation to such as the Linde process of liquefaction of gases, where it is very important to know what pressures to use to obtain the greatest cooling effect.

ZUSTANDGLEICHUNG DER DÄMPFE. Jar. Hybl. *Dinglers Polytechnisches Journal*, March 2, 9, 1912. 7 pp., 4 figs., 9 tables. *t.* Theoretical discussion of the *equation of steam and gases* ("steams"). The author denotes by the term "steams" gases which pass into liquid state at comparatively slight changes of temperature; each gas may be called steam when it is near its point of condensation, so that there is no distinct line of separation between gases and steams.

The author investigates the equations of state for superheated steam of Zeuner, Callendar, van der Waals, Tumlirz, and others, making use of data from the experiments of Mollier and Linde, and concludes that the most exact equations of state for superheated steam are those of van der Waals and Callendar, the second being more convenient for practical purposes owing to its comparative simplicity.

Besides steam, the gases most widely used technically in thermic processes are ammonia, carbon dioxide, and sulphurous acid.

For ammonia the author proposes the following variations of well-known equations for steam:

Tumlirz's

$$v = \frac{48.59\ T}{P} - 0.0137$$

Callendar's

$$v = \frac{49.7\ T}{P} - 0.0208 \left(\frac{273}{T}\right)^{2.22}$$

van der Waals's

$$P = \frac{49.7\ T}{v - 0.014} - \frac{461}{v^2}$$

and then shows that for saturated ammonia steam all equations, including those of Wobsa, give practically the same results. For superheated ammonia steam Callendar's and van der Waals's give again nearly equal values lying between those of the other equations

For carbon dioxide the author found, by comparing values obtained with various equations with the values from Amagat's tables, that all equations give values only approximately corrresponding to those of Amagat at low pressures, and give too high pressures at the critical point. The best results are obtained with Mollier's equation

$$P = \frac{19.32\ T}{v - 0.0002037} - \frac{19.36 \times e^{1 - \frac{T}{T_k}}}{(v + 0.0007719)^2}$$

where T_k is the critical temperature

$$T_k = 273 + 31.35 = 304.35 \text{ deg. cent. absolute}$$

and van der Waals's equation in the following form

$$P = \frac{19.3\ T}{v - 0.00027} - \frac{21.1}{(v + 0.00086)^2}$$

With superheated carbon dioxide vapor only the equations of Clausius and Mollier give values corresponding to those in the Amagat tables, while the other equations give more or less lower values. For high temperatures the equations of Clausius and Mollier give nearly equal values, the original equation of van der Waals higher, and his modified equation lower values, while the equations of Tumlirz and Callendar give considerably lower values.

For sulphurous acid the author found, on the basis of the tables of Cailletet and Mathias, the following forms of general equations of state

Tumlirz

$$v = \frac{13.51T}{P} - 0.01164$$

Callendar

$$v = \frac{13.3\,T}{P} - 0.008\left(\frac{273}{T}\right)^{-1.3}$$

van der Waals

$$P = \frac{13.3\,T}{v - 0.002} - \frac{40}{v^2}$$

For saturated vapor the Tumlirz equation gives values equal to those of the tables, while the other two equations give, for temperatures below 10 deg. cent., values slightly lower, and for temperature above 10 deg. cent., slightly higher than the values in the tables. The same is practically true for superheated vapor as well.

The article contains many tables and graphs comparing data from various tables with those obtained by the use of different formulae.

Miscellaneous

LA DÉCHÉANCE DES BREVETS D'INVENTIONS POUR DÉFAUT D'EXPLOITATION, L. Bidault des Chaumes. *Le Génie Civil*, March 9, 1912. 1 p. *gA*. Discussion of the *French law* of April 7, 1902, providing that a *patent for an invention* becomes *void if not worked* for two consecutive years in France, unless the patentee presents a good excuse for his inactivity. The French courts were formerly inclined to treat the provision of excuse very liberally: thus poverty of the inventor, or refusal of railroad companies to adopt an invention referring especially to railroads, were considered good excuses. Another tendency seems to have appeared lately: in the case of the Thermos Co. the court declared the patent void owing to inexploitation, notwithstanding the fact that the company has proved that it applied to several French glass factories, and that none was willing to undertake the manufacture of the Thermos botttles. The court held that the Thermos Co., to preserve its patent rights, ought to have established a special factory.

NEUE MÜLLEREIMASCHINEN, E. Redlich. *Der praktische Maschinen Konstrukteur*, March 14, 1912. 1½ pp., 3 figs. and 1 plate of drawings. *d*. Description of new flour milling machinery.

DAS MOTORPFLUGWESEN VOM STANDPUNKTE DER INDUSTRIE, Martiny. *Der Motorwagen*, March 20, 1912. 29 pp., 27 figs. *d*. Discussion of economic possibilities, and description of motor-driven plows, mainly those of the "Stock" and "Ihace" types. It appears that German motor-plow users have trouble because of the lack of technically skilled attendants. When, however, the author suggested at some meeting the advisability of sending some laborer to a motor-plow factory, where he could learn the construction and operation, he was told that that could not be done, "because the man won't come back," and he will not, says the author, as long as his wages are so low. The article contains also an extensive discussion of the question, in how far the use of motor plows can help the farmer to dispense with horses.

GAS POWER SECTION

PRELIMINARY REPORT OF LITERATURE COMMITTEE

(XVII)

ARTICLES IN PERIODICALS[1]

CAPACITY OF LARGE GAS ENGINES, INCREASING THE, F. E. Junge. *Power*, March 19, 1912. 4½ pp., 4 figs., 1 table, 3 curves.

The problem of increasing the output of large gas engines by scavenging by compressed air, and the difference in output of ordinary and scavenged engines.

COMBINATION POWER AND ICE PLANT, Paul C. Percy. *Power*, March 26, 1912. 3 pp., 8 figs., 1 table. *d*.

Describes a composite plant in which both steam and gas are motive powers and the products are ice and electricity. Gives results of a ten-day gas power run with producers burning wood scrap.

DIESEL OIL ENGINE AND ITS INDUSTRIAL IMPORTANCE, THE, Rudolph Diesel. *The Engineer* (London), March 22, 29, 1912. 6 pp., 24 figs., 1 curve. *cdhpA*. (To be continued.) Also *Engineering*, March 22, 1912. 11 pp., 22 figs., 1 table, 3 curves. *cdh*.

Discussion of the general importance of the Diesel engine and the questions brought up by J. F. Schubeler in a paper before the Zurich meeting of the Institution of Mechanical Engineers. Describes the different types and cycles of operation of Diesel engines, their adaptation to different classes of service, and economy of operation.

DIESEL MOTOR-DRIVEN LINER, SUCCESS OF THE FIRST LARGE, J. Rendell Wilson. *International Marine Engineering*, April, 1912. 6 pp., 8 figs.

General description of the engines and trials of S. S. Selandia of the East Asiatic Co.

FUEL IN THE GAS PRODUCER, SMALL-SIZED. *The Iron Age*, March 14, 1912. 2 pp., 2 figs., 1 table. *cd*.

Test results with the Kerply high-pressure type showing an advance in the use of low-grade fuels.

INTERNAL-COMBUSTION ENGINES FOR GERMAN FISHING BOATS. F. Romberg. *Engineering*, March 1, 1912. 3 pp., 29 figs.

Abstracts of paper before the Schiffbautechnische Gesellschaft, Berlin, November 23, 1911.

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[1] Opinions expressed are those of the reviewer, not of the Society. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *h* historical; *m* mathematical; *p* practical. A rating is occasionally given by the reviewer, as *A*, *B*, *C*. The first installment was given in The Journal for May, 1910.

LISTER-BRUSTON AUTOMATIC ELECTRIC LIGHTING PLANT, THE. *Engineering*, March 22, 1912. 1⅓ pp., 4 figs. *d.*

Describes a self contained petrol engine and dynamo for lighting country houses.

MARINE INTERNAL-COMBUSTION ENGINE, AN ANALYSIS OF THE CLAIMS OF THE. *The Engineer* (London), March 15, 1912. 2 pp., 8 figs., 1 table. *pA.*

Compared with steam engines.

OIL ENGINE DRIVEN WATERWORKS PUMP. *The Engineer* (London), March 8, 1912. ½ p., 1 fig., 1 curve. *mpB.*

Test of pumps and engine.

OIL POWER YACHT "MAIRI," THE. *Power*, April 9, 1912. 3 pp., 6 figs., 1 curve. *d.*

Describes the power plant consisting of a medium pressure reversing oil engine of the Diesel type.

PETROL-DRIVEN RAILWAY CAR. *The Engineer* (London), March 29, 1912. 1 p., 2 figs. *dpC.*

Passenger and mail car for the Midland Great Western Railway of Ireland with 26-h.p. vertical engine, 4 in. bore, 5 in. stroke, 1000 r.p.m.

PRODUCER GAS-DRIVEN CARGO VESSEL, SEA-GOING, F. C. Coleman. *International Marine Engineering*, April, 1912. 3 pp., 1 fig.

General description of and performance of cargo boat.

REVERSING GAS AND OIL ENGINES, A. M. Levin. *Power*, April 2, 1912. 4 pp., 5 figs., 1 curve. *d.*

Features of valve and reversing gears now in use on the smaller types of engines.

TURBINE, THE GAS, Norman Davey. *The Engineer* (London), March 8, 22, 1912. 4 pp., 4 figs., 2 curves. *dhmA.* To be continued.

Theoretical principles of the gas turbine, the air turbine, air turbine with external combustion, the closed cycle air turbine, the open cycle, internal-combustion in which the expansion takes place above the atmosphere, and internal-combustion turbines in which the expansion takes place below the atmosphere.

TWIN SCREW MOTOR SHIP SELANDIA, DESCRIPTION OF THE. *The Engineer* (London), March 8, 15, 22, 1912. 7 pp., 8 figs., one 2-page plate. *dpA.*

Size of ship, 370 ft. long by 73 ft. beam and 7400 ton capacity; engines: 1250-h.p., 8-cycle, 4-cylinder, Diesel type.

REPORTS OF MEETINGS

ENGINEERS DINNER AT PROVIDENCE

The Providence Association of Mechanical Engineers, affiliated with the Society, gave a dinner on March 26 in the Narragansett Hotel, which was attended by more than a hundred guests. Prof. Ira N. Hollis of Harvard University, senior vice-president of the Society, represented Dr. Humphreys, who was unable to attend, and extended the greetings of the membership and Council.

Professor Hollis was followed by Calvin W. Rice, Secretary of the Society, who spoke of the Society's activities and of its increased helpfulness and usefulness, by which it desired to benefit not only the membership but, in a modest way, the profession at large. He urged the participation of engineers generally in public affairs where engineering was involved and pointed out the especial need for development of foreign commerce. The balance of trade with South America is against the United States by over $100,000,000 gold annually, and is increasing from year to year.

Mr. E. L. Corthell, the speaker of the evening, was then introduced and made an eloquent address on the Future of South America, where he had spent many years of his life, giving many personal reminiscences of his work for the governments of South America, including the building of jetties, levees, and railroads. In conclusion, he announced a gift to Brown University, of which he is an alumnus, of his entire engineering library of over 7000 volumes, all bound in half morocco, and of provision for the continuance in perpetuity of subscriptions to the proceedings of all the 39 learned and engineering societies, both in America and abroad, of which he is a member.

PHILADELPHIA MEETING, MARCH 30

At a meeting of the Society in Philadelphia on March 30, a paper was read on an Ideal Flour Mill by B. W. Dedrick, instructor in milling engineering, Pennsylvania State College. This meeting was held at a time when there was a board meeting in Philadelphia of the Pennsylvania Millers' Association, so that the Society had the benefit of the discussion of those in attendance at this meeting. In the afternoon a visit was made to the Milburn Mills, Philadelphia, one of the oldest in the country. An abstract of the paper is published in this issue of The Journal.

SAN FRANCISCO MEETING, APRIL 3

At a meeting of the Society in San Francisco on April 3, a paper upon the Design and Mechanical Features of the California Gold Dredge, by

R. E. Cranston, was discussed. An account of the meeting will be published in an early issue of The Journal.

ST. LOUIS MEETING, APRIL 13

At a meeting on April 13 in the Chapel of the Second Baptist Church, held under the auspices of the Society, the Associated Engineering Societies of St. Louis coöperating, Dr. Rudolph Diesel of Munich, inventor of the Diesel oil engine, gave an illustrated lecture on the Development of the Diesel Engine, before an audience of over 500.

NEW HAVEN MEETING, APRIL 17

A meeting of the Society was held in New Haven, Conn., on the afternoon and evening of April 17, Manufacturing Costs being the general topic for consideration. The plan of a two-session meeting was tried by the New Haven members at a meeting last November, with such satisfactory results that this second meeting followed the same methods. The registration showed an attendance of 134 at the professional sessions held in the Mason Laboratory of the Sheffield Scientific School, and more than sixty were entertained at dinner at six o'clock at the Yale Dining Club. Considerable interest was manifested in the exhibition of some Gridley automatic turret lathes now in operation in the laboratory through the coöperation of the Windsor Machine Company.

The papers read at the afternoon session were The Development of Manufacturing Costs, by Prof. J. W. Roe, Mem.Am.Soc.M.E., assistant at Sheffield Scientific School, and Manufacturing Costs, by Bruce Fenn of Sargent & Company, New Haven. These were discussed by W. S. Huson, Mem.Am.Soc.M.E., general superintendent of the Whitlock Printing Press Manufacturing Company, Derby, Conn.; George W. Mixter, Mem.Am.Soc. M.E., vice-president of Deere & Company, Moline, Ill., and E. S. Cooley, Mem.Am.Soc.M.E., supervisor of power plants, N.Y., N.H. & H.R.R. Company, New Haven.

At the evening session a paper on the Cost Department and Its Relation to the Management, by G. P. Miller, secretary of the Bridgeport Brass Company, Bridgeport, Conn., was discussed by C. T. Raymond, Bridgeport, Conn.; E. W. Pelton, New Britain, Conn.; Arthur Brewer, Mem.Am.Soc. M.E., superintendent of the plant of the Bridgeport Brass Company; E. J. Mehren, New York; J. W. Roe, Mem.Am.Soc.M.E., New Haven; R. T. Kent, Junior Am.Soc.M.E., editor of Industrial Engineering, New York; M. W. Judge, Waterbury, Conn.; P. B. Stanley, New Britain; G. A. Kilborn, New Haven; Chas. F. Scott, Mem.Am.Soc.M.E., professor of electrical engineering, Yale University.

PROCEEDINGS OF THIRD NATIONAL CONSERVATION CONGRESS

The Executive Committee of the National Conservation Congress has decided to throw the reserve supply of the official addresses and proceedings of the Third National Conservation Congress open to the public. These are the only publications that record authoritatively the advance of the great conservation movement. They contain the speeches of President Taft.

Colonel Roosevelt, Gifford Pinchot, James R. Garfield, and other men of national reputation, as well as the reports of the state conservation commissions and of the conservation committees of national associations. As reference books on conservation they are invaluable.

As soon as the present limited supply is exhausted no more can be had at any price. While the supply lasts the complete set will be sent for $3, or $1 a volume, prepaid. On account of the limited supply it is probable that only early orders can be filled. Orders may be addressed to Thomas R. Shipp, Executive Secretary, National Conservation Congress, Indianapolis, Ind., with checks made payable to D. A. Latchaw, Treasurer.

STUDENT BRANCHES

ARMOUR INSTITUTE OF TECHNOLOGY

At a meeting held April 9 by the Armour Institute of Technology Student Branch the following officers were elected for the coming year: chairman, E. R. Burley; vice-chairman, J. D. Bradford; secretary, H. R. Kuehn; treasurer, A. Robertson. A paper on Oil Engines was also read by J. C. Miller and discussed by I. Newman and A. J. Beerbaum. The adaptability of the oil engine for power purposes was treated with regard to its reliability and the cost of operation and maintenance. Some of the most successful oil engines, such as the Diesel and De La Vergne types, were described in detail with the aid of a large number of lantern slides.

COLUMBIA UNIVERSITY

The Student Branch of Columbia University held a meeting March 29, at which P. Wood read a paper on Power Steering Gears on Modern Steamships.

On April 12, two papers were presented: Layout for a Manufacturing Plant, by B. Rogowski; and Precision Measuring Instruments, by B. Emmert. Discussion was offered by Messrs. Eddison, Thurston, Brombacker and Demorest. Professor Rautenstrauch and Messrs. Parr, Thurston and Herrick also gave short talks.

CORNELL UNIVERSITY

C. C. Anthony, Cornell 1886, assistant signal engineer of the Pennsylvania Railroad and son of Professor Anthony who founded at Cornell University the first course in electrical engineering in this country, spoke on Railway Signaling for Intensive Operation before a meeting of the Sibley College Student Branch April 11.

On March 15 Calvin W. Rice, Secretary Am.Soc.M.E., spoke on the Advantage of Membership in Student Sections of the Engineering Societies, and on March 29 J. C. Bishop read an original paper on Cement Manufacturing. The paper was commented on by Prof. R. C. Carpenter. A general discussion followed.

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

On March 19 the Mechanical Engineering Society of the Massachusetts Institute of Technology held its annual banquet and election of officers at

the Boston City Club. The officers are as follows: chairman, J. G. Russell; vice-chairman, H. D. Peck; secretary, J. B. Farwell; treasurer, L. L. Custer; governing committee, E. W. Brewster, W. H. Brotherton and M. L. Waterman. The speakers of the evening were Ira N. Hollis, I. E. Moultrop, R. E. Curtis, R. H. Rice and Prof. E. F. Miller, all members of the Society, and H. W. Hayward.

On April 3 the civil and mechanical engineering societies were addressed by James W. Nelson on High-Pressure Hydrostatic Machinery. After the discussion of many problems of construction and the manner in which they were overcome, lantern slides were shown, among which were some excellent pictures of the excavations and locks at Panama.

PENNSYLVANIA STATE COLLEGE

During his visit to the Pennsylvania State College on March 20, Calvin W. Rice, secretary of the Society, delivered two lectures, one to the entire engineering school when he dealt with the benefits to be derived from all engineering societies by its members, and another to the members of the mechanical engineering society, when he spoke more specifically concerning membership in the Society.

POLYTECHNIC INSTITUTE OF BROOKLYN

Edward A. Uehling read a paper on CO_2 Recorders before a meeting of the Polytechnic Institute Student Branch on March 30. Other speakers of the evening were Fred R. Low, F. W. Atkinson, G. A. Orrok, J. B. Chittenden and W. D. Ennis.

PURDUE UNIVERSITY

At a meeting of the Purdue University Student Branch on March 20, Prof. H. C. Peffer of the School of Chemical Engineering, spoke on Refractory Materials.

The meeting of April 3 was addressed by W. I. Battin of the Indiana Lighting Company, who spoke on The Manufacture and Distribution of Gas.

UNIVERSITY OF ARKANSAS

At a meeting of the Student Branch of the University of Arkansas held March 26, R. E. Thornton read a paper on Over-Speculation and Prof. W. N. Gladson read one on the Motor in the Machine Shop.

UNIVERSITY OF CINCINNATI

The University of Cincinnati Student Branch held an open meeting March 21, at which a lecture on Gears, Their Manufacture and Design, was given by Ralph E. Flanders, Assoc.Am.Soc.M.E. The discussion was led by John T. Rowell.

A meeting of the members was held March 26, and consisted of reviews of various current engineering periodicals by those in attendance.

UNIVERSITY OF ILLINOIS

The University of Illinois Student Branch held a meeting March 29, at which various methods of welding and their applications were discussed in papers by the following: Arc Welding, J. N. Todd; Flame Welding, A. L. Myers; Thermit Welding, A. H. Agaard. A general discussion followed.

On April 12 the section was affiliated with the Associated Engineering Societies of the university, the individuality of the branch remaining intact. The program included papers on the Muncie Oil Engine, by R. C. Chestnutt; and the Diesel Engine, by H. F. Crooks. A general discussion followed.

UNIVERSITY OF KANSAS

The Geology of the Earth's Crust by Professor Twenhoffel was the subject of the meeting of the University of Kansas Student Branch, March 7.

UNIVERSITY OF MISSOURI

A paper on Water Power, illustrated by lantern slides, was presented by H. E. Weaver and A. E. Pierce at a meeting of the University of Missouri Student Branch on March 20.

WASHINGTON UNIVERSITY

John Hunter, Mem.Am.Soc.M.E., gave an illustrated lecture on Central Power Plant Operation and Economy, with special reference to the Ashley Street plant of the Union Electric Light and Power Company, before the Washington University Student Branch on March 28. A general discussion followed.

YALE UNIVERSITY

At a meeting of the Yale University Student Branch on March 25, F. E. Booth read a paper on Railway Signaling. The lecture was fully illustrated by lantern slides showing the interlocking system as used by the Union Switch and Signal Company in the New York subway as well as in the systems of large railways.

At the meeting held on April 12, Dr. Alex. C. Humphreys, President of the Society, delivered an address concerning the position of the technical graduate upon leaving college.

NECROLOGY

HENRY W. SPANGLER

Henry W. Spangler was born at Carlisle, Pa., January 18, 1858. He was graduated with high rank at the United States Naval Academy in the class with two others who have attained to prominent positions in the field of engineering education, namely, Ira N. Hollis, head of the engineering department at Harvard University, and M. E. Cooley, dean of the department of engineering at the University of Michigan.

Professor Spangler was assistant engineer in the United States Navy from 1878 to 1889, although for about half of that period he was connected on detached service with the faculty of the University of Pennsylvania, first as assistant professor of mechanical engineering from 1881 to 1884, and from 1887 to 1889, and then as full professor, holding the Whitney professorship of dynamical engineering. During the Spanish-American War he served for a brief period as chief engineer in the United States Navy. With that exception he was in the service of the University of Pennsylvania as head of the mechanical and electrical engineering department continuously from 1887 till his death.

He was the author of a number of standard textbooks and numerous technical papers and professional reports. The textbooks from his pen embrace Valve Gears; Notes on Thermodynamics; Elements of Steam Engineering, which he wrote in collaboration with A. M. Greene and S. M. Marshall; Graphics; and Applied Thermodynamics. He was a member of the American Society of Naval Architects and Marine Engineers, the American Society of Naval Engineers, the American Society for Testing Materials, and the Society for the Promotion of Engineering Education, the Franklin Institute and the Engineers Club of Philadelphia. He was a member of the Advisory Council of the Engineering Congress of the World's Columbian Exposition in 1893 and of the Jury of Awards at the Buffalo Exposition in 1901.

In 1896 the University of Pennsylvania conferred upon him the

honorary degree of Master of Science and ten years later the degree of Doctor of Science. He died March 17, 1912, at his home in Philadelphia.

Edgar Marburg, professor of civil engineering at the University of Pennsylvania, pays him the following tribute:

As a writer, his chief characteristics were, perhaps, his painstaking efforts to present the subject in the simplest and clearest manner consistent with the intended scope of treatment, and to keep in view the practical requirements of prospective engineers, rather than theorists.

As a teacher, he was lucid, stimulating, progressive, and always intensely practical. His first concern was to help the students to gain a firm grasp of the underlying principles of the subject, and then to encourage them to rely on their own resources in the application of these principles. On no point, perhaps, was he more insistent than that of individual responsibility, which his students were required to assume in every branch of their work.

A strict and almost military disciplinarian, he was no less rigid in the standards which he applied to himself. The respect and admiration in which he was held by his students ripened into affection as they came to see him at closer range. There were few graduates who failed to turn to him at some time for helpful counsel in the perplexities of later years, or who failed to accept it, even though it ran counter to their own promptings.

He possessed to a remarkable degree the faculty of perceiving clearly, and almost intuitively, the essential elements of a seemingly difficult problem or complex situation, and he was as quick in action as in perception. Few excelled him in the clear discernment of the fallacies of an argument or in the directness of the challenge of such fallacies. Of a thoroughly progressive bent, he did not allow himself to be beguiled into strange paths by the educational fads and follies of the hour. The business of education was to him a serious business, with which liberties were not to be taken lightly.

EDWARD S. RENWICK

Edward S. Renwick, who died March 19, 1912, was born January 3, 1823, in New York, and when thirteen years of age entered Columbia University, the youngest member of his class. After graduation in 1839 he engaged in the manufacture of iron with the New Jersey Iron Company at Boonton, as assistant and bookkeeper to the superintendent. In 1844 he was employed to examine and report upon some mines in Maryland and was afterward sent to England to attend to matters appertaining to them. Here he had the opportunity to examine the best iron works in both England and Wales. In the fall of 1845 he returned to America and became superintendent of the Wyoming Iron Works at Wilkes-Barre, Pa., which comprised a merchant mill for rolling bars, a sheet mill, a mill for rods and hoops and a nail factory. He also put up a small blast furnace in the same town and engaged in the manufacture of pig iron. In the spring of

1849 Mr. Renwick went to Washington and associated himself with Peter H. Watson, subsequently Assistant Secretary of War under Stanton during the Lincoln adminstration, in the business of solicitor of patents and expert in patent causes in the United States Courts. While at Washington he made several inventions, the most important of which was the original self-binding reaping machine, which cut the grain, gathered it into gavels, compressed it, and bound it into sheaves, with wire, but preferably with twine. It remained dormant, however, until the seventies, after the expiration of the patents, when the impossibility of obtaining sufficient laborers to harvest grain in the old way forced it into use. This invention, with small improvements, has yielded millions to the makers of reaping machines, and has added untold wealth to the country.

In December 1854 Mr. Renwick returned to New York, and his first employment was as consulting engineer to Harrison Gray Dyer, who had become president pro tem of the New Haven Railroad. His employment in this position terminated with the resignation of Mr. Dyer and he resumed the practice of his profession as patent expert and consulting engineer. The most important engineering matter in which he was called upon to act was the repair of the Great Eastern steamer while afloat. This consisted in covering with iron plating a fracture of the bilge 27 feet beneath the water, 28 feet long and 10 feet 6 inches broad at its widest part, a feat declared impossible by other experts. He did the work in conjunction with his brother, Henry B. Renwick.

From 1865 Mr. Renwick's employment as expert in patent causes was continuous. He had probably been subjected to the longest cross-examinations of any expert. Thus in one of his early cases his cross-examination lasted twenty-one days, with the result that the United States Circuit Court and afterward the United States Supreme Court on appeal, adopted the construction of the patent given by him in favor of the defendant. In another case the cross-examination by the complainant's counsel was carried on in sessions of from one to three days separated by intervals, the whole amounting to twenty-two days running through six months, when the case was compromised for $15. In the Leffel waterwheel case he was cross-examined thirty-five days, the decision of the court being in favor of the plaintiff on whose behalf he had been called as expert.

Mr. Renwick was greatly interested in artificial incubation, and it was due largely to his efforts and inventions that the raising of

young chickens was rendered a paying industry. He was also responsible for many other inventions, among which are the balanced compound steam engine, and the system of surrounding that portion of the shaft of a twin propeller which extends beyond the vessel with a casing of sufficient size to permit this portion of the shaft to be inspected to the stern bearing. Both of these are described in his English Patent of 1868. The former invention did not come into use until the last decade of the last century; and the latter until the accident to one of the shafts of the City of Paris, since which it has been employed in all the great sea-going steamers with twin screws. In 1893 Mr. Renwick published his work on Patentable Inventions, which has been called the only sensible treatise ever written on the subject.

He was a member of the American Chemical Society, the Engineers Club, the Union Club, the New York Yacht Club, the St. Nicholas Society and other similar organizations.

LEE D. FISHER

Lee D. Fisher was born December 22, 1875, at Elyria, Ohio, and died February 8, 1912, at his home in Joliet, Ill. In 1897 he was graduated from Washington University as an electrical and mechanical engineer and accepted a position as boiler and elevator inspector for the Union Casualty and Surety Company of St. Louis. Four months later he was transferred to the company's New York office. When the Spanish-American War broke out he was appointed chief assistant engineer on the flagship New York and later was transferred to the transport Buffalo in charge of the engine room, making a trip to Manila, where he was temporarily assigned to shore duty as engineer in charge of fortifications during the insurrection of Aguinaldo. He returned to the United States on the Buffalo as chief engineer.

Mr. Fisher joined his father in 1900 in electric railroad building out of Columbus, Ohio, serving as chief engineer in the construction of three roads radiating from that city. In 1903 he aided in building the Joliet, Plainfield and Aurora Railroad and the Joliet and Southern Traction Company's line. In 1911 he accepted service with the Public Service Company of Northern Illinois, having charge of franchise matters for the company until last October, when he was made assistant to the vice-president and placed in charge of the publicity department of that corporation.

EMPLOYMENT BULLETIN

The Society considers it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for the Bulletin must be in hand before the 12th of the month. The list of men available is made up of members of the Society, and these are on file in the Society office, together with names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

0160 Works superintendent for plant employing 350 men, manufacturing heavy machinery, Corliss engines, etc. Superintendent must be competent to take full charge of entire plant. Apply through Am. Soc. M. E.

0161 Technical graduate having practical experience as an executive in charge of engineering department of a manufacturing plant, or as assistant to such an executive, wanted by Ohio concern. Familiarity with modern systems of management, construction of industrial and mining cars, design and construction of commercial auto trucks.

0162 Engineer experienced in automobile engine design and owning patent on silent sliding-valve engine which he believes has superior merit over best types now known, desires to interest capital to develop same.

0163 Mechanical draftsman experienced in the design of motors, generators and detail apparatus, such as controllers, switchboards, circuit breakers, etc. State experience and salary expected. Only first class men need apply. (Address Chief Clerk, Engineering Department, Westinghouse Elec. and Mfg. Co., E. P ttsburgh, Pa.)

0164 Mechanical engineer to collaborate with inventor in commercial development of a new cycle engine and later to manage the manufacturing end of the business. Man somewhat experienced n thermodynamics and ambitious to assist in promoting to the fullest extent the development of the mechanical end of the business, so that one would be preferred who can take a pecuniary interest in the business.

0165 Manufacturing executive for concern in New England, manufacturing electrical hardware, switches, cutouts, etc., small growing concern well supplied financially; executive with brains, experience and tireless energy. Salary $5000 upward according to man. Apply through Am. Soc. M. E.

MEN AVAILABLE

406 Works manager, long experience on light manufacturing involving in-

terchangeable parts. Competent to organize all departments of manufacturing plant along modern lines.

407 Mechanical engineer, age 34, technical graduate, four years' experience in design and construction of steam engines, two years' superintendent of engine works, three years' construction and complete power plant installation; desires position with sales, construction or manufacturing concern.

408 Mechanical and electrical engineer; graduate University of Pennsylvania, desires position in connection with sales department. Practical experience covers sales, mechanical design, charge of drafting room, foreman in machine shop and individual work on various machine tools. Detailed information and references given upon request.

409 Junior member of the Society, holding degrees of S.B. and M.M.E., four years' teaching experience and two years' engineering practice; desires position on instructing staff of mechanical department of some college.

410 Junior member, technical graduate, age 26, married, experienced in machine shop, automatic machine design, miscellaneous engineering work; now employed, but desires to locate with a reliable concern with opportunity to advance. Can furnish the best of references of character, ability, reliability, etc.

411 Graduate Mass. Inst. Tech., 25 years of age, experienced in drafting and designing of special machinery and buildings for foundry, machine shop, rolling mill and special manufacturers, shop experience and in the manufacture of steel office, bank and library furniture and floors. Desires position in the mechanical engineering line in or near Spokane, preferably that of assistant to manager or superintendent.

412 Position as manager of small growing factory desired by mechanical engineer, 32 years old, technical graduate, three years' experience as machinist apprentice, two years' drafting, four years' production and industrial engineering installing shop and cost systems, rearranging and equipping plants for increased efficiency. Familiar with laying out new plants and electrical installations. At present general superintendent of factory employing 150 to 200 men. In charge of manufacturing, production, factory accounting, and costs.

413 Member, with extensive experience in large machine and tool manufacturing plants, finest and most accurate classes of precision work to the heaviest; thorough practical knowledge of tools, machinery, equipment, modern shop practice organization and management, seeks connection as superintendent, master mechanic or factory manager where there is real need of man who can do things. Salary not less than $3000 and can earn more.

414 Junior, technical graduate, 32 years old, married, ten years' practical experience, mostly steel mills. At present district sales manager in Middle West; desires change requiring less traveling. Would like to interest party with capital in complete engineering sales line for Birmingham District.

415 Member, experienced in design, construction, operation and testing, gas and oil engines, steam engines, compressors, pumps and general power plant work; good shop man. Formerly chief engineer for old established firm; now

connected with one of largest universities in the country. Desires to secure position from June 1 to October 1.

416 Member, mechanical engineer, Cornell graduate, having 12 years' experience in designing automobile, stationary and marine internal-combustion engines; desires position.

417 Member, graduate mechanical engineer, with extensive experience in manufacturing interchangeable machinery. Specially good experience in the design of machinery, jigs, and fixtures for the production of duplicate parts, also in testing, handling and use of glues and adhesives. Present position mechanical engineer with large wood and iron working factory. Wishes position as superintendent.

418 Technical graduate with 18 years' experience in shop, drafting room, office and teaching in mechanical engineering, the last including executive work, would like to change. Desires to become connected with consulting engineer or with engineering department in college or university.

419 Member, technical education and wide experience in different kinds of mills, steam engineering and modern machine shop practice, desires position as superintendent, master mechanic or chief engineer of power plant.

420 Young mechanical engineer, technical graduate, experienced along lines of building steam piping, design and erection, desires position offering chance of advancement with engineering firm engaged in similar work. Good references.

ACCESSIONS TO THE LIBRARY

With Comments by the Librarian

This list includes only accessions to the library of this Society. Lists of accessions to the libraries of the A. I. E. E. and A. I. M. E. can be secured on request from Calvin W. Rice, Secretary Am. Soc. M. E.

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